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THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

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The LOCAL ORGANIZATIONS OF THE SOCIETY

THAT no man liveth to himself alone is a truth well taught by the developments of modern science, and now forced home to us as a matter of life and death in carrying on the war. The members of the Society must do their utmost, therefore, to cultivate the habit of engineering cooperation, of giving that they may receive, if democracy is to continue its forward movement.

The local organizations or Sections are a most potent factor in giving the automotive engineering assistance now required by the Government. Each individual member of the Society, wherever he may be, is inspired by his contact with the Society to greater effort for the cause. Two members in a town can exert more than twice their individual effort; similarly, when a large number of members are united in a Section, the total effort for the common good is greater than the sum of what can be accomplished by the individual members working alone. Thus the work of the Sections, situated as they are in all the important centers of the automotive industry throughout the country, must be continued with even more vigor than during times of peace.

New Sections must be formed in many localities, wherever the number of members or those eligible as members is sufficient to carry on the work. The conduct of a Section requires the hearty support of all the members in its territory. It requires Section officers who will lead the work along broad and constructive channels.

The Society is now, and has been in the past, exceedingly fortunate in the officers of its Sections. The present officers are all men who have taken up the Section work in spite of heavy burdens in their daily business life. They are doing a patriotic service in thus voluntarily adding to their tasks. They deserve the highest credit for it. There is an opportunity for members in many other places to establish Sections that will increase the effectiveness of the Society as a factor in helping to win the war.

At present there are only eight Sections of the Society. The Buffalo Section consists of members located in the western part of New York State; the Cleveland Section of those in northern Ohio; the Detroit Section draws its members from that city and surrounding territory; the Indiana Section from all over that State; the Metropolitan Section from the city of New York and surrounding cities; the Mid-West Section from

Chicago and cities near it; the Minneapolis Section centers in Minneapolis and St. Paul; and the Pennsylvania Section covers the State.

Although these Sections carry on their work according to rules approved by the Council of the Society, they are practically self-governing. Their control is in the hands of officers elected by the section members themselves, but these officers, of course, are in close touch with the Council of the Society and with Society Headquarters at New York. This contact enables the local Section officers to promote the national solidity and unity that are essential in carrying on the Society work.

The main activities of the Sections are the presentation of technical papers. During the last year these have been mainly of a war nature, relating to the design, construction, maintenance and operation of different types of war automotive apparatus. The Minneapolis Section, in its first year, has had presented a notable series of papers on tractor engineering subjects. Other Sections have had papers on tractor, aeronautic, and on all the various automotive engineering subjects. These papers, together with the discussions accompanying them, have been printed in THE JOURNAL of the Society, and have been reprinted in a large number of technical and trade periodicals in this country and in foreign countries. The wide distribution of the technical papers presented before the Sections is characteristic of the importance of the work they are doing. The gathering of 100 engineers in Minneapolis may result in the publication of facts that will solve problems being encountered by hundreds of others in different parts of the country. The Section meetings are of benefit not only to those in attendance, but also to many others, who can read the reports of the meetings.

In addition to the production of papers, one of the great activities of the Section is the promotion of good-fellowship. This has been done by the acquaintance-work at the meetings, by the dinners held previous to the meetings, and by visits of inspection, or outings.

The Pennsylvania Section recently, for instance, conducted a successful trip to Hog Island, where the members inspected the immense shipbuilding developments. The outing was concluded by a dinner held at a Philadelphia hotel. The Metropolitan Section is now planning for a trip to a military camp near New York. The work done during the past year by the Mid-West Section is a good illustration of how a section can cooperate in carrying on Society activities. Committees of this

Section conceived the details and made all the arrangements for the great War Dinner of the Society, held in February, at Chicago. In the early part of June the Section conducted a successful joint technical meeting and dinner with the National Gas Engine Association at the time of the annual convention of the latter.

A very important function of the Section is its help in securing new members for the Society and consequently for itself. This is carried on through membership committees appointed by the Sections, and cooperating with the Membership Committee of the parent Society. A good example of this is the work done by the Cleveland Section during the 1917 drive for membership, when a large and enthusiastic membership committee, which took charge of the membership activities of the whole state of Ohio, was successful in securing many new members for the Society.

The Constitution of the Society permits its Council to authorize the organization of new sections, which

The exact method to be followed will depend upon local conditions to a large extent. On the Atlantic Coast there are possibilities for sections in Massachusetts, with headquarters at Boston, and including the surrounding cities as far as Worcester; in Connecticut, with headquarters perhaps in Hartford, and including the members in New Haven, Bridgeport, Springfield, New Britain and Bristol. In New York State a Rochester branch might be formed to work with the Buffalo Section. A section in Syracuse could work with the Technology Club there. The city of Washington, in which 135 members are now located, offers a possibility for some sort of a local organization. Washington is, of course, a Society center now because of the office there. In the state of Pennsylvania most of the section work is now concentrated in Philadelphia, although Pittsburgh has a large number of members who should be organized. There is room for several sections in the state of Ohio. Dayton has nearly 75 members, and

OFFICERS AND COMMITTEE CHAIRMEN OF S. A. E. SECTION ORGANIZATIONS

Name of Section	Place of Meeting*	OFFICERS						CHAIRMEN OF REGULAR SECTION COMMITTEES		
		Chairman	Vice-Chairman	Treasurer	Secretary	Address of Secretary	Member Governing Committee	Meetings	Membership	Entertainment
Buffalo	Hotel Statler	H. R. Corse	David Fergusson	O. M. Burkhardt	E. T. Larkin	1252 Niagara St., Buffalo	John Younger	John T. R. Bell
Cleveland	Hotel Statler	H. H. Newsom	S. G. Thompson	R. S. Begg	R. E. Clingan	1036 Guardian Bldg., Cleveland	H. G. Welfare	H. E. Figgie	L. L. Williams
Detroit	Hotel Pontchartrain	J. E. Schipper	A. C. Hamilton	Don T. Hastings	C. F. VanSicklen	701 Book Bldg., Detroit	†	R. H. Sherry	G. W. Meridith	F. W. Watts
Indiana	Hotel Claypool	Earl Bessom	W. S. Reed	Geo. T. Briggs	Geo. T. Briggs	Shelby & Barth Sts., Indianapolis	†	E. Bessom	W. S. Reed	G. T. Briggs
Metropolitan	Automobile Club of America	C. F. Scott	H. G. McComb	A. C. Bergmann	C. E. Hunt	553 West 23d St., New York	H. A. Goddard	N. B. Pope	L. I. Stewart	E. Favary
Mid-West	Chicago Automobile Club	G. W. Smith	C. S. Whitney	Lon R. Smith	D. S. Hatch	2105 Mallery Bldg., Chicago	G. L. Lavery	F. W. Parker, Jr.	F. Parker, Jr.
Minneapolis	Hotel Radisson	H. C. Buffington	J. L. Mowry	J. S. Clapper	C. T. Stevens	139 So. 9th St., Minneapolis	†	A. W. Scarratt	R. B. Shoop
Pennsylvania	Engineers Club of Philadelphia	C. A. Musselman	J. W. Watron	W. M. Newkirk	H. E. Rice	4937 Stenton Ave., Philadelphia	T. Y. Olsen	H. R. Cobleigh	J. A. Anglada	H. E. Rice

*The meeting places are subject to change in some cases; those given are the latest definitely known.

†In these Sections the Chairmen of regular Section Committees are members of the Governing Committee.

may put into effect a Constitution and By-laws, provided these are in harmony with the Constitution and By-Laws of the National Society and are approved by the Council. The Council thoroughly recognizes the value of the Section work and encourages the formation of active local organizations. Information regarding the operating procedure of the Sections can be obtained from the New York Office of the Society by any members interested in forming a section.

The distribution of Society membership in the United States is indicated in the accompanying map, on which the name of every city with five or more members is printed. It would seem from this that a considerable number of new sections or branches of existing sections might be formed. In some cases these may well be State organizations, and in others sections formed to cooperate with local engineering bodies. Still another possibility is that branches of the existing sections may be formed, these to hold separate meetings and to meet also at times with the main body.

At the present time there seem to be three methods by which the Section activities can be extended: (1) By the formation of entirely independent sections; (2) by the formation of sections to cooperate with existing local engineering organizations; and (3) by the extension (as branches) of present sectional organizations to meet the needs of groups of members concentrated within their territory.

Akron, Cincinnati and Toledo are strong centers of S. A. E. membership. The Indiana Section activities are concentrated mainly in Indianapolis, although Muncie and Lafayette are rapidly developing into section centers.

The Mid-West Section is intended to cover the parts of Wisconsin, Illinois, Indiana and perhaps Michigan, within traveling distance of Chicago. There seem to be a number of opportunities for section organizations in this district. A "quad-city" section has been suggested by members from the cities of East Moline, Moline, Davenport, and Rock Island, which are very close together. Racine and Milwaukee, in the state of Wisconsin, are large centers of membership and possible section headquarters.

Going further West, St. Louis is rapidly increasing in number of members, and a section should be established in that city. On the Pacific Coast, San Francisco is the largest center of membership, there being over 25 in or near the city. The group of members at Los Angeles might well form a branch of a Pacific Coast Section.

Leaving the United States itself, we find that a considerable number of members are located in Canada, and the formation of a Canadian Section of the Society is a possibility. The Society now has a large number of members in England and France, most of whom are in or near the cities of London or Paris.



MAP SHOWING DISTRIBUTION OF S A E MEMBERS THROUGHOUT UNITED STATES. TOWNS IN WHICH
BY THE DOW ONLY. PRESENT SECTION AREAS AND POSSIBLE SECTION AREAS

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In addition, many American members have occasion to visit these cities, so that considerable thought is being given to the possibility of forming a French and an English Section of the Society. It is believed that these sections could be of real assistance in the present international emergency by the exchange of information with engineering organizations and governmental departments, and by serving as a foreign headquarters for members abroad. The desirability of forming local organizations is now being considered by the members in both England and France, and some action will probably be taken to secure closer affiliation among these foreign members and between them and the Society in this country.

SECTION CONFERENCE AT DAYTON

The variety of Section activities was well shown at the conference of Section officers held on the evening of the 16th of June, at Dayton. The conference was devoted mainly to the discussion of the methods of holding Section meetings and to the duties that should be undertaken by Section officers and committees. These methods and duties have been very completely outlined in a Section Manual, which is a reduction to words of a "standard practice" method of operating sections. The Manual was revised somewhat at the meeting, and will now be sent to all Section officers for their future guidance.

During the conference several of the different officers gave an outline of their plans for Section meetings during the coming year. These show that a great deal of careful thought has already been given the subject. For instance, Chairman C. F. Scott of the Metropolitan Section outlined the plans his section has formulated, these comprising the holding of meetings to be devoted to such broad subjects as highway transport work, tractor manufacturing possibilities, commercial aspects of aeronautics, rubber-tire development, and to war automotive activities. Since the Metropolitan Section is not in a purely manufacturing district it is considered desirable to hold meetings relating to general subjects, and not to specific phases of design. The Section plans to hold one meeting in cooperation with the Pennsylvania Section, this for the benefit of members between New York and Philadelphia who cannot get to either Section meeting very often. The joint meeting may be held in New Brunswick, or in some town near it in New Jersey.

Chairman J. E. Schipper described the plan followed by the Detroit Section, in which a large Meetings Committee is appointed; this is divided up into subcommittees, each of which is responsible for a meeting. This plan was very successful last year, and will probably be repeated the coming year. Mr. Schipper said that every effort must be made to give the members who attend Section meetings information in which they are directly interested. Papers must be presented with more life in them than ever before. Motion pictures have

been found very successful for Section meetings, particularly when they relate to the use of automotive apparatus in war.

Relations with the Public

Many of the Sections are taking an active part in public affairs related to automotive engineering. It was the opinion of the conference that Sections should be prepared to give advice on public matters involving automotive engineering questions, although it was not to be expected that they could handle any great amount of detail work. But it was the feeling that the Section members should appreciate their duties as citizens of the communities in which they live. One way in which a good deal of work can be done is through the local and state automobile clubs and associations. Messrs. Strickland of the Cleveland Section and Briggs of the Indiana Section said that they had found this method best for their respective localities.

Secretary Darwin S. Hatch of the Mid-West Section described the newly formed War Service Committee of the engineering societies of Chicago. Nineteen of the engineering societies organized locally in Chicago have representatives on this committee, the purpose of which is to enable the technical societies in the Chicago district to call into play the efforts of their members as occasion may arise, and to coordinate their activities into the most effectual channels to help win the war. Waldo G. Gernandt is the Mid-West Section representative on the War Committee.

Local Interest in Sections

In response to a question from Chairman H. R. Corse of the Buffalo Section as to whether representatives of local dealers and service men attended meetings, the fact was brought out that many of the papers are of interest to such men, and that they are glad to attend meetings of the Sections.

Mr. Hatch told of the movement now being furthered by the National Automobile Chamber of Commerce and the National Automobile Dealers' Association to instruct new service men and to make the present ones more efficient. He said that the Sections could do a patriotic service by devoting at least one meeting each year to the technical aspect of service work. Chairman H. C. Buffington of the Minneapolis Section stated that many garages throughout the country are looking forward to handling tractors, and for that reason any discussion as to the construction of tractors interests the people connected with the garages.

So many matters were brought up for discussion at the conference that there was not sufficient time to consider the proposed revisions of the By-Laws of the Sections, or to discuss a number of forms proposed for standardization, so that the "paper" work of the Section may be uniform. These will be sent to the Section officers, however, and a mail vote will be taken as to the desirability of accepting them.



Screw Thread Situation in Great Britain and America

By E. H. EHRLMAN* (*Member of the Society*)

SEMI-ANNUAL MEETING REPORT

Illustrated with CHARTS

IN order to intelligently understand the elementary features of screw threads with Sellers and Whitworth profiles, which will be discussed in this report, it is well to review briefly their early history.

Originally, screw threads were of two simple shapes, V and square; the latter found application in screws for transmitting energy (chiefly as one of the mechanical powers), while the former, being about twice the strength of the latter, and mechanically more easily produced, naturally has been used for constructional bolts.

The impossibility of maintaining a sharp point, especially with the early methods of thread cutting, either in the tool or on the screw, led naturally to developments along two lines. In the one the shape that the tool assumed through wear was accepted. This practice was supported by experience, in that the life of a tool with the point rounded was greatly increased. In the other development the source of the trouble was removed by truncating the tool point. Thus, there came into existence in Great Britain (1857) threads rounded at the crest and root, according to a profile advocated by Sir Joseph Whitworth, and in America, a few years later, threads in which the crest and root were flattened according to a formula recommended by William Sellers. Each system has since become standard in its respective country and its use has increased until it is quite general.

PROFILES NOT ARBITRARY

The round shape of the Whitworth profile was the result of study and experience. It was found that the wear of a V-thread tool-point, at first rapid, decreased as it became blunter until a stage was reached at which the shape might be called stable; that is, for a certain depth of thread Sir Joseph Whitworth adopted as a standard crest and root rounding the least round of tool-point that could be considered permanent.

In the bolt, the rounding both of the crest of the thread to overcome the liability of its being marred in handling and of the root to get added strength were marked improvements over the sharp V-thread, which was formerly common practice in England. Similarly the flattened crest and flattened root in the Sellers thread apparently gained the same ends and had, it was believed, the additional advantage of greater simplicity of construction.

In the early practice, particularly in the making of the larger sizes of bolts and taps, the thread proportions were obtained in a relatively simple manner. The workman ground the threading tool to the proper angle by hand as accurately as he could; in a similar manner the point of the tool was flattened or rounded by eye or to gage, and the thread on the bolt or tap cut to the proper diameter, measurements being made by caliper at the

root and crest. In the case of the Whitworth thread, it was necessary to shape the crest with a separate tool, this being done after the flanks and root were finished. In the case of the U. S. S. thread, the prior turning of the blank to the proper diameter gave the correct profile when the thread had been cut to the right depth. In the period above referred to, the means of making and gaging screws and taps were simple, and the accuracy of both manufacture and gaging was crude as compared with present practice.

Much of the gaging was done by mating the parts themselves, and shop interchangeability was built up around the tap. General interchangeability was obtainable only through relative looseness of fit, aided by the manufacture of taps sufficiently oversize to take care of individual shop variations. These means of securing interchangeability have obtained until recently, one of the last refinements resulting from the screw manufacturers agreeing to use one "make" of taps in tapping and testing their gages. It is quite possible that this practice still exists in some shops, even to this day.

The advance and refinements in machine manufacture have necessitated corresponding refinements in the manufacture of screw-threading tools and of screws and nuts, to maintain interchangeability as well as closer fits.

Closely associated with accuracy in screw manufacture is that of thread gage manufacture; so much so that the non-uniformity in gage limits is becoming a most serious matter, particularly when the errors of manufacture in the gage cannot be held below one-eighth the tolerance permitted in the manufacture of the product.

The above introduction might indicate that the next step naturally would be to consider which of the two standards might be "best." The details in profile, material, proportions, etc., are analyzed differently by various persons and committees, so that naturally the conclusions arrived at are different. If all could look at things in the same light and from the same viewpoint, all could agree on the one "best" system, and we might have a universal standard, not alone in the matter of screw threads, but in other important industrial and engineering matters as well.

This report of the conferences with the British Engineering Standards Committee on screw threads, and of the status of the Whitworth thread in Great Britain, will be best presented, I believe, by reviewing the several discussions and also the various practices obtaining in both countries.

The conference in London was called for the purpose of discussing the present screw thread standards, and if possible to recommend the adoption of a uniform system for English speaking nations. It was at once apparent that the difference in custom and practice of the two countries was such that a most careful study in each

*Secretary and Factory Manager, The Chicago Screw Company

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country by the representatives of the other country would be required before any recommendations could be made that might lead ultimately to such a millennium. With this thought in mind, the several details—among these profile, interchangeability, pitch, tolerance, limits and gage factors—will be analyzed briefly.

COMPARISONS OF THREAD PROFILE

The Whitworth profile, Fig. 1, for the purpose of comparison, is superimposed on the Sellers profile. The depth of the Whitworth thread is two-thirds the depth of a 55-deg. sharp V-thread while the depth of the Sellers thread is three-fourths the depth of a 60-deg. sharp V-thread. The depths of both profiles are practically alike, being $0.6403p$ and $0.6495p$ respectively. For $\frac{1}{4}$ -in. pitch, the difference in depth is but 0.002 in., and for $\frac{1}{8}$ -in. pitch it is but 0.001 inch. The rounded crest and root of the Whitworth thread reduce the useful bearing depth 23 per cent, making it $0.4906p$. Neglecting for the moment the slightly differing resultant pressures due to the $2\frac{1}{2}$ -deg. difference in slope of thread, the length of engagement of a bolt having the Sellers-profile thread need be but 75 per cent that of a bolt of the same pitch having the Whitworth-profile thread, both under the same tension, and the pressure per unit of area on the thread slopes being the same.

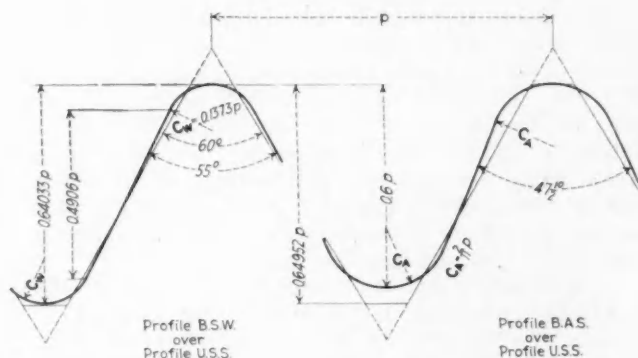


FIG. 1—(LEFT) BRITISH STANDARD WHITWORTH AND (RIGHT) BRITISH ASSOCIATION STANDARD THREAD PROFILES SUPERIMPOSED ON SELLERS THREAD PROFILE

The advantages of the shorter engagement are (1) less trouble because of error in lead, and (2) reduced weight if the engaging member is a nut.

The Whitworth and Sellers profiles cross or overlap owing to the slight difference in the thread angle. If, however, the profiles be separated radially about 0.047 of their depth, the interference ceases. This movement amounts to but 0.0015 in. for $\frac{1}{20}$ -in. pitch so that a nut of either profile 0.003 in. larger in pitch diameter than a screw of the other profile could be assembled with the latter. This would indicate that, for purposes of repair, nuts and bolts of the same nominal size and pitch, although not of the same profile, might be interchangeable.

Experiments to determine the possibilities along this line are being conducted in both countries. The National Physical Laboratory, through its director, Sir Richard Glazebrook, has received specimens of commercial U. S. S. screws and nuts and has also, at the author's request, furnished us with specimens of Whitworth bolts, together with a record of the measurements in detail. Specimens of U. S. S. bolts are now being measured at

the laboratory of the British Ministry of Munitions in New York. When this is done, the extent of their interchangeability with British bolts will be determined, and the findings compared with those obtained in England. It is expected that even where the two profiles may somewhat overlap and interfere, the slight flowing of the metal will permit their assembly with at most but a light wrench (spanner) pressure. The value of this commercial interchangeability is apparent as it provides, in the event of the superseding of one profile by the other, for preserving interchangeability during the period of transition.

The Whitworth profile has been a subject of so much study and discussion among our British friends, that it gives one the impression that, while correct from a theoretical viewpoint, somehow, simple as it appears, the profile presents practical problems that have proved troublesome and that still call for solution—problems that we believe our Sellers profile does not impose upon us. I refer to what I believe is the best and at the same time the worst feature of the Whitworth profile—the rounded crests and roots, which is best in service (may we grant for the moment) when true to shape, but often worst in service when this is not realized; withal I believe in many respects it is much more difficult of realization than is our own thread profile.

In support of the superiority of the Whitworth profile may be cited the deductions of C. E. Stromeyer, who in a very able paper* draws some remarkable conclusions; among them one, that under the conditions assumed the principal resultant stresses under the roots of threads are less for the Whitworth than for the Sellers profile, and under most conditions slightly greater for the coarser than for the finer pitches. Other interesting deductions are made regarding the comparative stresses at the thread roots in cases where bolts are used with nuts of different thicknesses, in which the pitch of the nut differs slightly from that of the bolt. The distribution of stress was found to be improved when the pitch is coarser in proper relation to the tension and to the elasticity of the metal, but aggravated if the nut pitch is finer. Some applications of this analysis are made by Mr. Stromeyer in connection with the failure of large piston rods and main bearing bolts in the threads. In both instances the pitch of the threads in the nuts was slightly less than that on the rods and bolts, thereby forming a combination conducive to excessive stresses.

Mr. Stromeyer refers to some interesting experiments made by L. Rowland,† in which sheet-celluloid nut and bolt sections were stressed; while under stress polarized light was passed through them, and photographic impressions made showing the stresses set up at the roots of the threads.

It would be interesting to learn from actual bolt and nut tests if the "flow" or "set" of the metal does not tend to offset to some extent the errors in thread profile and pitch so that the stresses are not so serious as the experiments with the celluloid model would indicate.

I have, for some time, had in mind conducting a series of tensile tests of bolts in which the pitch of the thread differed slightly from that of the engaging nuts, being in some instances finer and in other instances coarser—the difference of course amounting to but a few thousandths of an inch such as might creep in through carelessness in manufacture. Such a study would prove an

*Stress Distribution in Bolts and Nuts. Inst. of Naval Architects, March, 1918 meeting.

†Trans. Inst. of Naval Architects, Vol. LX, 1918

interesting sequel to the analysis above referred to, and particularly so if both U. S. S. and Whitworth threads were tried under similar conditions.

I have recently learned that Mark Barr, Esq., a member of the British Engineering Standards Committee, has suggested* a series of tests along somewhat similar lines. He proposes to test in varying combinations bolts and nuts in which the difference in thread angle may be from 0 to 5 deg. and in pitch from 0.000 to 0.003 inches. Naturally, if a parallel research is to be conducted in this country, the conditions attending the tests should be the same, with the possible exception of the thread profiles. The results of each research would then have added value. Under my direction a series of tests were recently made on threaded standard tensile specimens, identical in all respects except that some had Whitworth and some Sellers threads, both being of the same core diameter. There was far more variation in the tensile strength of specimens having the same thread profile than between the two sets having different profiles, indicating that when made of the material used in the test bolts of one thread profile would be as strong as bolts of the other.

An interesting conclusion drawn from the result of these tests is that the strength of a bolt is in excess of that computed upon the basis of the core area, by about 11 per cent.

Looking at the matter from the industrial viewpoint, one finds the question of clearance linked closely to crest and root problems, both in practice and in discussion.

Information obtained from several sources indicates that British practice in providing screw-thread clearances is extremely limited. However, a few of the most prominent companies do make a practice of providing clearances, some by cylindrically truncating the thread and some by deepening the thread. (See Fig. 2, F and C.)

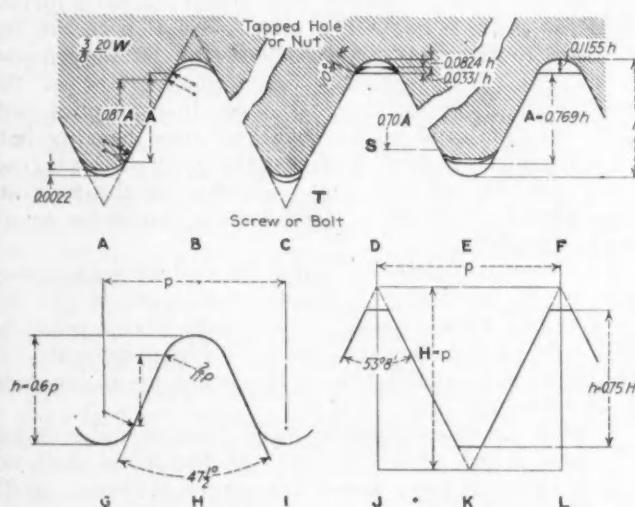


FIG. 2—ILLUSTRATING CLEARANCES PROVIDED BY DIFFERENT MANUFACTURERS

Wm. Taylor, Esq., in a clear analysis† of this subject, holds that it is futile to obtain clearance along the above or similar lines, as proposed by some manufacturers, be-

cause it is gained at the expense of strength in the core, thread bearing, or life of threading tools.

Fig. 2 shows the several methods of clearance, most of which are considered in Mr. Taylor's analysis.

A.-B.—Clearance of 0.0022 at crest of 20-pitch Whitworth thread; reduces depth of engagement 13 per cent.

C.—Clearance of 0.0022 at root of 20-pitch Whitworth thread; reduces strength of bolt through diminishing core diameter.

D.—Crest truncated to 10 deg. of its base. Gives crest clearance without entailing loss in depth of thread bearing, until the edges carried away by the threading tool drop below the base of the round.

E.—Same as F, except that the corners of the crest are rounded by being carried away by the threading tool. The profile just above, terminating at S, indicates the contour of the crest as affected by tolerance.

F.—Crest truncated to base of round.

The arguments cannot be better stated than in Mr. Taylor's paper, from which I quote as follows:

"In C. L. 3786, page 5, Mr. Sears, who has had great experience in measuring commercial screws, points out that we must have tolerances considerably larger than those specified in report No. 38, and he proposes, for example, in the case of the $\frac{3}{8}$ in. British standard fine (B. S. F.) screw, which has twenty threads per inch, to give a tolerance of 0.0022 in. on both the full diameter of bolt and the core diameter of the nut. The effect of such tolerance on the crests (disregarding any on the effective diameter and roots) is illustrated to scale in the accompanying figure (A, Fig. 2) in which the part cross hatched is the nut and the other part the bolt, the thread crests being reduced by such amount of tolerance. It will be seen that the depth of engagement C is about 13 per cent less than the nominal depth A. This, though an appreciable loss, appears inevitable.

"A proposal has been made to alter the Whitworth thread by truncating to cylindrical form the crests of the threads (F, Fig. 2), but it is recognized, even by the advocates of this change, that in practice we shall not realize this thread form, but that the crests will be rounded off somewhat in the manner of the dotted lines.

"If we apply Mr. Sears' tolerances to this truncated thread, the practical result will be that we shall cut down the depth of engagement to S, Fig. 2, which is about 30 per cent less than the nominal depth. This is a serious loss. If the crests were rounded with circular curves the loss would exceed 60 per cent.

"Thus, to truncate the Whitworth thread merely weakens it, without yielding any practical advantage. The idea seems to be like will-o'-the-wisp and as futile as a dog's chasing his tail.

Tolerance on the Roots

"Fig. 3 shows in outline the Whitworth Standard thread, and the form of the roots of the thread with the tolerance Mr. Sears suggests for the $\frac{3}{8}$ -in. B. S. F. screw. Tolerance at the root of the thread must always be an encroachment on the triangular space T (see Fig. 2, C), which is enclosed by the sides of the fundamental triangle and the root of the standard thread. We can term this the truncation area. The whole of this truncation area is not available for tolerance, however, because it is impossible in practice to maintain (or even to attain) really sharp

*Letter to British Engineering Standards Committee, Dec. 6, 1917, file C L 4011.

†Notes on Thread Angle and Tolerances. British Eng. Standards Committee report, C L 3888.

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points on the tools; and in the present example of a thread, 20 per inch, Mr. Sears' tolerance is from this point of view about the utmost that is practicable. As the pitch of thread becomes finer, that part of the truncation area which is available for tolerance shrinks more and more as a proportion of the whole. As Mr. Sears has shown, however, a *larger* proportion of tolerance is needed with such threads."

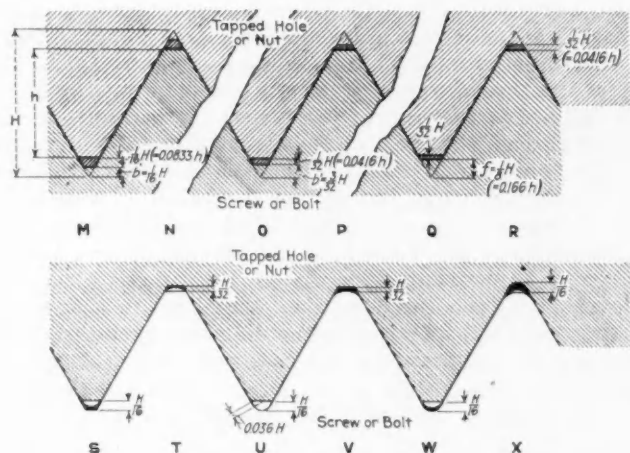


FIG. 3—FORM OF ROOTS AND TOLERANCES OF WHITWORTH STANDARD THREADS

Mr. Taylor also shows the difficulty in securing clearance through deepening the thread in the finer pitches in the International or Sellers systems, taking as an example a thread of 1-mm. pitch.

"This (speaking of a root with the tolerance proposed by Mr. Sears) falls almost at the apex of the truncation area, and as the radius of the *normal* root of this thread scarcely exceeds 0.002 in., and it is impracticable in any case to make the root sharper, it follows that this thread form, for such a pitch, gives no possibility of any tolerance at the root, at any rate tolerance which is plus at the root of the nut thread or minus at that of the bolt thread. It also follows that threads of the nominal International form, of still finer pitch, are impracticable of realization. And what is true of the International thread is generally true of the U. S. thread, except that there is even more difficulty with the latter because its truncation area is smaller, being limited at its base by a straight line.

"There are only two ways in which it is practicable with threads of U. S. and International form, and especially with the finer pitches, to realize adequate tolerance at the roots. One is to make the threads correspondingly free on their effective diameters, for this again enlarges the radius of the root, but this need not be further considered here because our present object is to secure closer fit on effective diameter by means of the increased tolerance at the roots.

"The other way is by taking what is commonly termed the clearance at the roots of these threads, and regarding it as a tolerance and not as a clearance. Owing, however, to the cylindrical form of the thread crest which determines the boundary surface over which the coating root must not step, and to the inevitable rounding of that root as in the International thread, there is not in this clearance area sufficient space available for root tolerance. As may be seen (W, Fig. 3) the space avail-

able corresponds to less than half of Mr. Sears' proposed tolerance."

Mr. Taylor shows clearly that to attain in the finer pitches a root clearance of any practical value, the thread angle must be more acute than in either the Sellers or Whitworth profile. This is due to the increase in the depth of the truncation area as the thread angle becomes more acute.

The reason for this is apparent, as the clearance profile below the root will, for a given depth of clearance, have a radius larger in proportion as the depth of the truncated area is greater.

On account of Mr. Taylor's prominence in the industry, his arguments command attention.

Why we in this country have not experienced the troubles cited in his arguments, I can only surmise. It is possible we do have them, but they may not be properly diagnosed as such.

No doubt the crests of our threading tools become rounded in use, and especially when made so as to produce root clearance.

This would be most pronounced in the manufacture of rough bolts and nuts, where in quantity production, and with cheap threading lubricant, the tools are expected to have "nine lives."

There would, however, be one consolation if the situation is as bad as has been apprehended; namely, that with its sharp corners rounded off, the Sellers thread would have one less disadvantage as compared with the Whitworth profile.

Again if a study of this detail should develop the fact that the rounding of the corners assumed a definite magnitude by the time the wear of the rest of the profile of the threading tool (tap or die) made it unfit for further use, the extent of rounding might be the basis for the depth of the clearance space.

There is little doubt as to the advisability of providing clearance between the crests and roots of thread. The tendency is to provide therefor by a suitable tolerance and proper limits, one of the limits being the basic crest or root. It is on this basis that the screw thread specification proposed by the A. S. M. E. provides for root clearances.

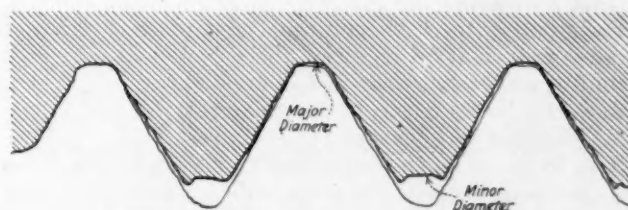


FIG. 4—THREAD PROFILE ON A 50 TO 1 SCALE
Scale reduced in reproduction to about 15.4 to 1

I have hoped for a long time that the clearance might be obtained entirely through modification of the nut thread; such a result is secured at the slight expense of only one detail—depth of thread bearing, and this is not in any sense a sacrifice, as now in ordinary practice the nut bore is often greater than required for necessary clearance. A most important feature with this method is that the clearances can be completely provided for in the tap, and so can be made use of or not as occasion may require, without entailing any change in the threads of the bolts, screws or studs.

Through the courtesy of the director of the New York Laboratory of the British Ministry of Munitions, thread profiles on a 50 to 1 scale have been made of several

sizes of U. S. S. bolts, one of which is shown in Fig. 4. This furnishes a most interesting study in its several details, which are briefly summarized:

The bore of the nut is drilled larger than the theoretical core diameter; the spinning effect of the tap threads slopes is shown in the projections at the edges of the nut thread-crest.

The roots of the nut thread indicate that the tap crests are rounded but very slightly at the corners.

The crests of the screw thread do not show much evidence of having the corners torn away by the die.

The root of the screw thread might indicate (1) uneven wear in the die crests; (2) deformation of the die crests at one corner owing to a piece from a fractured crest near the throat of the die becoming embedded in the bottom of the screw thread and rubbing against the die crests as they passed over it; and (3) original defect in die (result of faulty workmanship).

This profile suggests that when the threading tool does not have to remove any metal from the crest of the thread in the product there is a tendency, at least under some conditions, to raise the thread slightly at times by spinning. Might it not be possible to find, by trial, a position for the truncated crest, *D*, Fig. 2, such that a sufficient tolerance clearance could be provided without the loss of any depth of engagement?

For the purpose of comparison, the Sellers profile is shown with the same pitch as in the Whitworth and British Association profiles; several types of clearance are shown, together with wear in the tool thread crests permissible before interference would result in the threads of the product. Referring to Fig. 3:

M-N.—Clearance (see *U*) A. S. M. E. machine-screw standard.

O-P.—Root clearance equals one-fourth depth of truncation area; in use to some extent in the United States and specified by some Government departments.

Q-R.—Clearances—nut crest and root (see page 125).

S.—Shaded portion shows extent of wear in crest of tool producing root clearance *M* before interference with engaging thread crest (perfect) would interfere.

T.—Similar to *S* except for tool crest *O*.

U.—International System (metric) preferred contour of root clearance.

V.—Similar to *T* except that advantage is taken of possible deformation of screw thread crest at corners to permit of additional wear in tool crest.

W.—Similar to *S* except for tool crest *U*.

X.—Similar to *V* except for tool crest *U*.

It is to be noted in concluding this part of the report that clearance between the crest and root provides a space into which the metal can flow when the fit is wrench tight.

As a matter of interest in connection with thread profiles, the other important profiles are illustrated:

Fig. 2.—*G-H-I*, British Association Standard.

Fig. 2.—*J-K-L*, Loewenherz (German).

The way several minor details are affected by the profile will now be described:

Ease of Production

The elements of the Whitworth profile are much more closely linked together than those of the Sellers profile, so that a change in any element affects the adjoining elements. The troublesome link is the *tangency* of the slope to the arcs at the crest and root; almost any change or correction in the slopes of the thread necessitates a cor-

rection in the radius of the crest or root, or in both to re-establish depth and tangency. In the Sellers profile the same change would necessitate but one other correction—to re-establish the width of flat at the root. This interdependence of the thread elements has a most important bearing in tool and gage making, especially in the latter, since it affects the facility of attaining accuracy and economy.

Influence in Manufacturing

In cutting Whitworth threads (with tap, die, or chaser), there is a decided tendency to tear the thread at the crest because of the inability of the metal at the top of the thread to withstand the stress of cutting along the full arc at the root of the thread in the tool. In the Sellers thread, the roots of the thread of the tool have no cutting to do (if the bolt, nut or hole has been machined within proper limits before the threading is done) except to remove the slight burr that may be thrown up owing to the flow of the metal in threading.

Again, adjustment of a die of Whitworth profile to correct the pitch, or effective, diameter of the screw thread being cut affects also the diameter over the crest, whereas this need not be the case with a die of Sellers' profile.

Use of Fine Pitches

The difference in the viewpoints, in England and in the United States, so far as concerns pitches finer than Whitworth or U. S. S., is most interesting, likewise the difference in experience.

The standard Whitworth pitches agree essentially with those of the United States and if no other pitches were in use in either country the way that leads to a unification of the two systems would be clear and the end of the road soon reached. But there are other series of pitches in both countries, forming so to speak a fork in the road; whether either country can retrace its steps to the juncture and proceed down the fork taken by the other country is a question, but it must be done unless an additional series of pitches is also standardized by one country.

In England the situation as I understand it is this: with advances and refinement in industry, the need was felt for a system of pitches finer than Whitworth. It is quite probable that the use of fine pitches in the early bicycle days had an influence and hastened the realization of the present B. S. F. The B. S. F. system, however, is not a standard supplemental to the B. S. W., forming within the latter a dual system. I believe I am correct in stating that the B. S. F. is intended to supersede the B. S. W.; it is not too fine to meet all requirements heretofore imposed by the latter, and at the same time not too coarse to be adapted to the needs of the automobile and kindred industries.

In our own country, the need of a system of finer pitches has been met by the S. A. E. standard. The U. S. S. series has not been superseded, but it and the S. A. E. make a dual system meeting a wide range of requirements.

The pitches in the three systems are as follows:

Diameter	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	1 $\frac{1}{4}$	1 $\frac{1}{2}$
B. S. W. & U. S. S.	20	18	16	14	12 (13)	12	11	10	9	8	7	6
B. S. F.	26	22	20	18	16	16	14	12	11	10	9	8
S. A. E.	28	24	24	20	20	18	18	16	14	14	12	12

In England the belief is common that the S. A. E. series is unnecessarily fine, finer than need be to meet requirements, and furthermore, to its fineness of pitch

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are attributed the difficulties experienced in the assembly of bolts. In other words, the danger of "crossing" the threads in starting a nut on a bolt or a screw into a tapped hole apparently increases with the fineness of the pitch, and a workman who is none too careful may (and the experience abroad goes to show he has) by faulty starting on of nuts unknowingly ruin the threads on the bolts, thereby producing conditions that may lead to serious consequences.

There appears to be a united stand taken on this point so that undoubtedly it is based on experience and not alone on theory.

One can only conjecture why—at least so far as my personal experience goes—we have been free from similar troubles in this country, and also whether there may be underlying causes other than merely the fineness of pitch.

One member of our commission has suggested that unless removed the partial first half thread on the bolt, or in the nut, might contribute largely to the trouble. True it is that the first quarter thread if not removed can in handling be pressed into the groove immediately behind it and cause serious trouble by partly blocking the groove; true it is too—and this is borne out by personal observation—that the practice of *not* chamfering the first thread in either internally or externally threaded work is altogether too common in England. In this country, too, we are far from being "not guilty" as to this practice.

I do not believe that the above is the only contributing cause; it is not unlikely that inaccuracies in the rounding of the crests and roots of the thread not revealed by the method of gaging, together with too meager tolerances, and allowances, or lack of clearance, might cause interference of crest with root, with the result that the threads easily become crossed in assembling.

It is not impossible that the fundamental contour or profile of the thread might be involved unless safeguarded by sufficient clearance.

Since my return, repeated inquiry has failed to produce evidence of troubles and difficulties developing through the use of S. A. E. pitches. I believe that data on this subject should be obtained, possibly through a questionnaire sent not only to manufacturers in the automotive industries, but also to those in other industries, including especially the machine tool makers, so that we can learn if possible the true status of the situation.

The S. A. E. series appeals to me as having good features, among them:

- (1) Increased core diameter; hence greater strength.
- (2) Closer adjustment with the same angular movement of nut; this is especially valuable, since the nut can be locked at regular angular intervals.
- (3) Greater facility and better quality in manufacture, especially when the material is alloy steel.
- (4) Though the depth of thread is less, the depth of bearing is from 3 to 23 per cent greater than for the same sizes in the B. S. F. system, except for screws larger than 11/16 in. outside diameter.

I hope that some research work may be done with special reference to the stresses below the thread roots, covering a range that will include S. A. E. nut proportions, the materials used in both nut and bolt, and the stresses the bolt is designed to withstand. The results of such a study would be a valuable check on the design of thread profile and crest and root clearances. I am confident the findings would confirm those* of the com-

mittee that recommended proportions for aircraft bolts and nuts a year or more ago.

The objection that S. A. E. pitches are too fine is advanced by some members of the B. E. S. A., and in support of this they say that we confine their use to bolts, and when screws or studs engage holes tapped in metals softer than steel, it is necessary to use U. S. S. pitches. Reliable data regarding present practices and the reasons therefor will be required to answer this objection.

Almost immediately upon my return home, I studied the drawings from a number of representative manufacturers in the automotive industry in order to determine the trend in practice, at least in a general way.

The fact that engineers of high standing in the industry consistently use S. A. E. pitches for entire units, such as engines, indicates at least that they are adaptable to and serviceable in metals other than steel.

May not the use of coarser pitches than the S. A. E. be due to reasons entirely aside from the inapplicability of the latter? For example, the practice of others may be followed; coarse pitches may be retained through mere apprehension that finer pitches are not suitable; the proper proportions of the tapped hole in non-ferrous metal may be lacking; lack of refinement in manufacture, and failure to provide proper clearances.

I cannot believe that mere fineness of pitch, of itself, prevents the use of S. A. E. pitches in non-ferrous metals, and I should regret to be forced to admit that I am mistaken.

Disposition of Limits

Two important factors, (1) the degree of interchangeability and (2) the quality of fit or play, are dependent upon the disposition of limits.

To insure interchangeability without sacrificing closeness of fit, either the minimum hole size or the maximum screw size must be fixed. For several good reasons it is better to fix the screw size than the hole size.

The practice in Great Britain and that in the United States on this point are fortunately in accord, thus affording one less obstacle to international interchangeability, and also establishing a precedent based on sound principles. With the maximum, or high, screw limit thus fixed (at the nominal size), for all except wrench or force fits, both allowances and tolerances can be kept as narrow as manufacturing practice will permit, without affecting interchangeability.

The French practice, as explained at the conference, is to fix the minimum hole, or tap size, and to secure the allowance desired by varying the maximum screw size.

Even if the same system of measurement were standard in France and the English-speaking countries, interchangeability would not be insured in all cases. In one case, a force fit might result and in the other case great looseness.

TOLERANCES AND ALLOWANCES

Tolerance affects facility of production and in conjunction with allowance determines the quality of fit.

Allowance is the specification factor that determines the minimum play, or looseness of fit; tolerance and allowance together determine the maximum play or looseness of fit.

The British definitions of these two terms are in substance as follows:

Allowance is the difference in dimensions (of the maximum screw and the minimum nut) prescribed in order to allow for various qualities of fit.

Tolerance is a difference in dimensions [of the maxi-

*Subcommittee on Screws and Bolts of Aeronautic Division of the Standards Committee, composed of C. B. King, C. M. Manly, F. G. Diffin and E. H. Ehrman

imum screw (or nut) and the minimum screw (or nut)] prescribed in order to tolerate unavoidable imperfections in workmanship.

Specifications of screw thread tolerances and allowances are usually based on formulas made up of functions of the pitch or the diameter.

In the British Report No. 38, dealing with limits and tolerances, the formulas for tolerances on the full and core diameter and the tolerance are functions of the square root of the diameter (\sqrt{D}); that for the tolerance in pitch is a function of the fourth root of the diameter ($\sqrt[4]{D}$) and that for the tolerance on the effective diameter is a function of the fourth root of the diameter cubed ($\sqrt[4]{D^3}$).

In the proposed B. S. F. screw standard, report No. C. L. 4369, which supersedes the B. S. F. standard in report No. 38, the formulas for all diameter tolerances are functions of the square root of the pitch (\sqrt{p}), thus being on a basis different from that used in the earlier standard.

In the A. S. M. E. machine-screw standard, all diameter tolerances are functions of the *threads per inch*.

In the specification proposed by the A. S. M. E. for medium-fit screw tolerances, the formulas for tolerances on the external and root diameters are functions of the *threads per inch*, while those for lead and pitch diameter tolerances are functions of the square root of the diameter (\sqrt{D}).

In the report (4369) on B. S. F. screw threads, above referred to, the errors provided for in the pitch diameter tolerance are analyzed. The term "grade" is used to denote the quality of workmanship, while the term "play" denotes the shake in or on a theoretically accurate "Go" gage. To illustrate: assume that a screw is smaller than standard by 0.003 in.; if the pitch and slope angle be perfect the screw will have a shake or "play" of 0.003 in. in a standard gage; if, however, there are errors of pitch, or angle, or both, such that the total magnitude thereof, measured by the effect in pitch diameter, is 0.002 in., the play would be reduced to 0.001 in.; if the effect of these errors were 0.003 in. there would be no play.

The grade of a screw is determined by the total of all the errors—those of pitch diameter and of pitch and angle measured by the alteration in pitch diameter to compensate therefor. In the example just given, the tolerance specified, 0.003 in., permits of errors in pitch and angle affecting the pitch diameter a total of 0.003 in.; hence the grade is six.

In this report due consideration has been given such factors as pitch, angle, and crest and root errors, and provision made therefor in the proposed specification of tolerances. This is apparent upon comparison with those contained in report No. 38, the new tolerances in pitch diameter being from 50 to 100 per cent greater than those in the old standard; the tolerances in the full and core diameters are liberal in comparison with those of older standards.

The appendix of this report (4369) contains analyses of pitch and angle errors, together with a chart by Capt. P. Bishop for determining the "grade" and "play" graphically.

In the new A. S. M. E. specification the influence of pitch errors is analyzed along a different line from that pursued in the British report; in the former, the tolerance on the pitch diameter is that required to compensate for a prescribed pitch error in a length of engagement of one diameter; in the latter the prescribed pitch error is

not expected to be exceeded in the length of engagement (which is not specified, except in a general way as that to be found in "ordinary nuts and bolts").

In the A. S. M. E. specification, as in the British specification, the tolerances at the root and crest are such that a clearance is provided.

The crest tolerances approximate those given in the British report, but those for the root provide for much less clearance. It is my belief that such factors as strength, bearing surface and tool wear must be carefully studied before the clearance area can be standardized; the remarks on this detail in the earlier part of this report indicate the attention it is receiving in England.

In the new British Specification, the allowance is the same for all sizes, being 0.002 in.; in the A. S. M. E. specification the tap-screw allowance amounts to one-fourth of the screw tolerance; the tapped hole-screw allowance varies from this amount to nothing.

The inference should not be drawn that a zero allowance means a wrench (spanner) fit. Several factors tend to make the actual fit wholly different from the "specification" fit; among these are (1) the law of averages (including variations in diameter, pitch or lead, slope angle, etc.); (2) the fact that the fit between the check and the gage is closer than that between the screw (or nut) and the gage, which in turn is closer than that between the screw and the nut (or tapped hole) on account of the relatively irregular surfaces in the product, the gage coming into contact with the "high spots" only; (3) the laying over or packing down of the surface imperfections produced by the threading tools; and (4) the packing, or flow of metal due to the stresses set up in use.

This difference between "specified" and "actual" tolerance could often be taken advantage of, thereby permitting the manufacturer more reasonable manufacturing tolerances without lowering the quality of fit; the gain would increase in production, decrease in rejections; additional cost—nothing. This subject is treated in detail under "gages."

EFFECT OF LEAD (OR PITCH) ERRORS

The effect of progressive error in lead (pitch) measured by the compensation in pitch diameter required, is a function of the cotangent of half the thread angle, consequently the more acute the thread angle the greater the reduction to compensate for an error in lead.

When Z equals the difference in lead of the mating threads, for the length of thread engagement, this compensation in pitch diameter is as follows:

Sellers thread	1.732Z
Whitworth thread	1.921Z
Thury thread	2.273Z

Thus for a given pitch error the alteration in pitch diameter to compensate therefor is, in the Whitworth thread, almost 11 per cent, and in the Thury (or B. A.) thread, fully 30 per cent more than in the Sellers thread.

For a given error per inch, in lead, the pitch diameter is affected thereby in direct proportion to the length of engagement. If the lead in a tapped hole differs from that in a screw, the bearing, at first between the two threads at one end of the length of engagement, extends under stress, axially, progressively to the other threads, as the metal yields by compression, flow or distortion. The changes permissible in the screw threads to co-ordinate the lead errors without materially affecting the serviceability of the assembly determines the magnitude of difference of lead errors allowable.

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In practice a slight difference in lead (in the proper direction) between tapped hole and screw under tension is preferable to no difference in lead. A slightly plus pitch in the nut (or hole) will shorten through compression, while a slightly minus pitch in the screw will lengthen through tension, owing to the stresses, the amounts being dependent on the elasticity of the materials and the stresses imposed. The difference in pitch diminishes and under a certain stress is practically nothing—so that, as far as the pitch is concerned, each thread comes nearer bearing its share of the total load under these conditions than is possible when the pitch in the mating parts is identical.

Reversing the signs of the lead errors creates a condition the seriousness of which increases more rapidly than the difference in lead error. This point is clearly brought out in Mr. Stromeyer's paper, to which reference has already been made.

This case is mentioned partly to emphasize the importance of closely watching errors of lead, and partly to show that, in the inspection of bolts and nuts (or taps) the direction of trend of the lead error should be considered. In the new British specification, half of the tolerance compensates for error in lead, the remaining half compensating for error in angle, but it is not contemplated that either error will be such as to require the total tolerance specified.

If this practice is strictly followed, the net tolerance available to offset lead error averages about the same as in the new A. S. M. E. specification, in which with the small angular error permissible only a small part of the total tolerance is required to compensate for it.

Effect of Errors of Angle

In the new British screw thread specification an interesting analysis is made of errors of angle and the provision therefor that must be made in the pitch diameter tolerances. The influence of error in angle is greater in the Sellers profile than in the Whitworth, as it increases with the depth of engagement.

This is evident from the formulas:

$$y = 0.0105p[(\alpha \curvearrowright 27\frac{1}{2}) + (\beta \curvearrowright 27\frac{1}{2})] \text{ Whitworth thread}$$

$$y = 0.01309p[(\alpha \curvearrowright 30) + (\beta \curvearrowright 30)] \text{ Sellers thread}$$

in which y = the reduction in pitch diameter to compensate for errors in the slope angle.

α = angle in degrees between a line normal to the axis and the slope of one side of thread, measured in the axial plane.

β = angle in degrees between a line normal to the axis and the slope of the other side of the thread, measured in the axial plane.

\curvearrowright signifies difference between.

As stated under "Effect of Lead Error," the British specification deals with an angle error requiring the same amount of tolerance in effective diameter to compensate for it as does error in lead or pitch, whereas the A. S. M. E. specification deals with it as an error of much less importance owing to its better initial control in the threading tool.

This is evidenced by the result of some recent measurements made of specimens of nuts and bolts of from $\frac{1}{4}$ to $\frac{5}{8}$ -in. diameter. In a lot of 12 nuts the error of angle in eight nuts was 0 deg.; in three nuts, +1 deg.; and in one nut, -1 deg. Similarly in a lot of 12 bolts the angular error in six bolts was 0 deg.; in two bolts, 2 deg.; and in two bolts, 5 deg. A notation in the re-

port stated that in the last instances the threads were poor (torn), indicating that the result might be due to the material being none too well suited for machining purposes. This inspection test, while it can not be construed as representative, might indicate that the A. S. M. E. angle tolerances, although suitable as to their application to threading tools, may hardly be sufficient to cover errors in product of the grade for which the specifications are intended.

The angle tolerances in the A. S. M. E. specification do, however, rob the tolerance provided to compensate for pitch error from 18 per cent for the smallest size to 12 per cent for the largest size; it is likely that the average actual angle errors would probably be such as to rob the specified tolerance of double these amounts.

Experience shows that the sides of the threading tool (tap or die) wear most appreciably near the point. The angle between the thread slopes in the product has a tendency, therefore, to become slightly more obtuse. Within the constructional bolt ranges this tendency is more pronounced on external threads than in internal threads, as angle errors caused by tool wear lie in the same direction; the net result does not, however, disturb the quality of fit materially, in comparison with the results of the other thread errors.

The permissible errors as given in the British specification range from 5.2 deg. total for 28 threads to 2.1 deg. total for five threads per inch. While these angle errors seem large, they should not be compared with the A. S. M. E. permissible angle errors for reasons intimated above and also on account of the differences in the make-up of the Whitworth and Sellers profiles.

APPLICATION OF GAGES

The complete measurement of a screw comprises measurements of:

- (1) The full (outside, or major) diameter.
- (2) The pitch (or effective) diameter.
- (3) The core (minor) diameter.
- (4) The angle the slope makes with the axis (or a line normal thereto).
- (5) The pitch (or lead).
- (6) Rotundity.
- (7) Parallelism (and taper).
- (8) Correctness of profile.

Some of these measurements, such as those for angle, diameter at the root, rotundity, parallelism and correctness of profile, need not be made so frequently as those for the remaining dimensions, as these points are primarily affected by the tooling and machine conditions and so are not subject to fluctuations to the same extent as are the outside and pitch diameters. The lead is so important that it is classed with the diameters, as a point requiring close attention in the gaging of screw product.

For most threaded product (not in the "precision" class) two (or at most three) gages are needed:

- (1) "Go" gage.
- (2) "Not go" gage.
- (3) Pitch (or lead) gage.

The "Go" gage is usually the counterpart of the piece to be gaged, and, in the words of a British authority on screw gages, "Must gage simultaneously all the elements concerned in order to insure their proper correlation."

This requirement makes screw gages expensive. All of the eight points noted above have to be taken into account within a very few ten thousandths of an inch.

In gages having threads of Whitworth profile, the

maintenance of the shape at crest and root makes them more difficult to manufacture than are gages having Sellers threads. In the latter, that part of the gage that checks the thread diameter at the crest may be made an auxiliary gage. This subdivision of the gage permits the finishing to size of the slopes, crest and root independently, without in any way affecting the other elements.

The auxiliary gage is often used separately in gaging; this procedure does not, however, insure in the part being gaged concentricity of the crest, with the slopes and root.

The subdivision of the gage having Sellers thread reduces the cost to from one-half to one-third that of the complete form gage with Whitworth thread.

The "Not-go" gage is also the counterpart of the piece to be gaged, but gages only the pitch diameter. The crest and root should therefore be cleared and the length of the gage need not be greater than that of a few threads.

The pitch gage is used often as an auxiliary to the "Go" gage when the latter indicates the existence of appreciable error in pitch.

When depth of thread engagement, and strength of bolt require checking, "Not-go" gages that measure the crest and root diameters are also necessary.

In England the practice, to a considerable extent, has been to use plain plug, ring or caliper gages as "Not-go" gages. With this type of gage a piece might be passed in, on which the thread was so "thin" as to be properly rejected by a threaded "Not-go" gage.

It must not be understood from this comment that "Not-go" checking is limited to this method; in fact, the recommended practice is that in which "Not-go" gages are used for all three diameters.

Gage Tolerances and Limits

In England, the principle that the manufacturer be allowed the full tolerance that exists between the limits specified for the product, is well established; the same principle applies in the same manner to the limits of gages, the product of the gage maker.

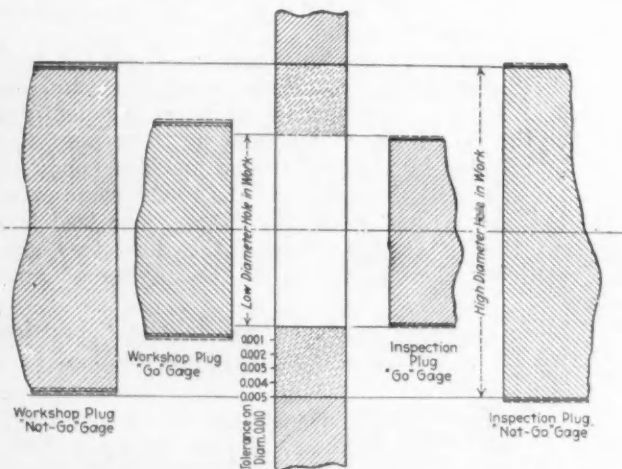


FIG. 5—LIMITS ON FINISHED WORK AND ON INSPECTION GAGES

The British National Physical Laboratory is very explicit on this point. The following is taken from its pamphlet on the subject of gages:

"Munition work is officially checked by means of 'Go' and 'Not-go' inspection gages, made within such limits

that the gages do not encroach on the tolerance nominally allowed for the work. As a simple example, work in the form of a plain cylindrical ring may be allotted to a tolerance from 1.490 to 1.500 in., in which case it would be inspected by a 'Go' plug gage, diameter 1.490 in., or slightly smaller, and a 'Not-go' plug gage, diameter 1.500 in., or slightly larger.

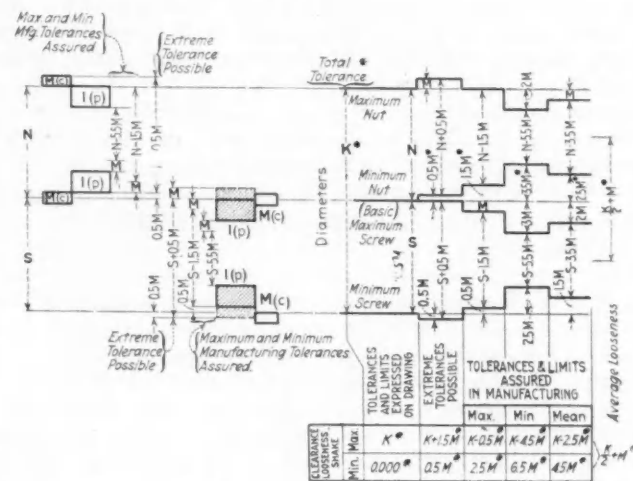


FIG. 6—SIMPLE SCHEME OF THREAD GAGE TOLERANCES

"Since a number of nominally identical gages cannot be made absolutely alike in size, and as they are also subject to wear in use, gages made to the same limits as official inspection gages should not be used to govern work in the shop. If gages used in the shop are made to inspection limits, work they pass during manufacture may subsequently be rightly rejected by the official inspection gages. This cannot be too strongly emphasized. Fig. 5 shows diagrammatically the limits on the finished work and on the inspection gages, together with suggested limits for workshop gages; the limits suggested in the diagram for workshop 'Go' gages are arranged to allow for some wear during use.

"Referring to Fig. 5, it will be seen that all work made within the nominal limits will pass the official inspection gages; and further, owing to the necessary tolerance allowed on the inspection gages, some work will pass which is slightly outside its nominal limits. The larger the tolerances on the inspection gages, however, the more widely may some of the work passed by them fall outside the nominal limits allowed for it."

In our country this policy is far from being general. The lack of uniformity makes the import of a specification of tolerance a matter of doubt; construed in one way, the full tolerance is for the use of the manufacturer, to dispose of as he may see fit between the product and the working gages; construed in the other way, deductions must be made, often for as many as four gages other than those he is interested in as working gages. Gage tolerances, although specified, may with difficulty be attainable, so that although the prescribed gage accuracy is technically within his control he is often a loser on account of the gage makers, on whom he must usually depend, not being able to produce the gages within the limits allotted. The result is smaller net tolerances than the manufacturer had counted on.

One of the simplest schemes of gage disposition is illustrated in Fig. 6. The customer's gages number three—all external or plug check gages, the tolerances

SCREW THREAD SITUATION

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of the "Not-go" plugs being so disposed that their limits do not fall within the limits of the product. In the "Go" gage the limits + and - are equal. The net results of this disposition of gage tolerances is as follows:

Extreme tolerance possible $S + 0.5 M$ and $N + 0.5 M$
 Max. tolerance assured... $S - 1.5 M$ and $N - 1.5 M$
 Average tolerance $S - 3.5 M$ and $N - 3.5 M$
 Mean max. looseness of fit. $N + S - 2.5 M = K - 2.5 M$
 Mean min. looseness of fit. $4.5 M$
 Average looseness of fit... $\frac{1}{2}N + \frac{1}{2}S + M = \frac{1}{2}K + M$

This scheme was, last January, suggested by the author to one of the Government departments where the closest fits possible without overlapping of limits were desired, but without reducing the manufacturing tolerance to such an extent as to make them impracticable.

The practice in Great Britain, as I understand it, is shown on Fig. 7. It will be noted that the inspection gage tolerances are entirely outside the product limits, and that the inspection gage check limits do not encroach on the inspection gage tolerances. Thus, the only deductions the manufacturer must make from the prescribed tolerances for the product are the tolerances for his own inspection gages.

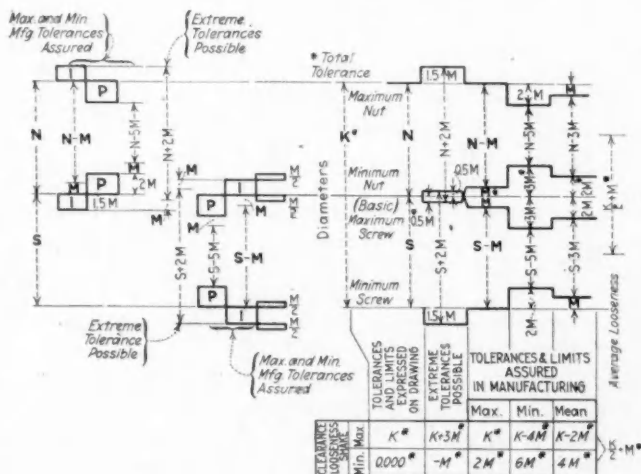


FIG. 7—BRITISH PRACTICE IN THREAD GAGE TOLERANCES

In marked contrast to the above scheme is that in force to some extent in this country, in which all gage tolerances are deducted from the tolerances for the product. Fig. 8 shows graphically the shrinkage in net tolerances due to this scheme.

Below is a summary of the net tolerances, etc., obtained with each of the three schemes (Figs. 6, 7 and 8) for disposition of gage limits:

	Fig. 6	Fig. 7	Fig. 8
Specified tolerance...	N	N	N
Mean net tolerance...	$N - 3.5 M$	$N - 3 M$	$N - 8 M$
Specified allowance*	None	None	Z
Mean net allowance.	$4.5 M$	$4 M$	$9 M + Z$
Apparent mean looseness (specification)	$\frac{1}{2}K$	$\frac{1}{2}K$	$\frac{1}{2}K + Z$
Net average looseness	$\frac{1}{2}K + M$	$\frac{1}{2}K + M$	$\frac{1}{2}K + M + Z$

*Neutral zone, minimum play or shake

The last item (looseness) is subject to a possible error of $0.5 M$, an allowance made for the difference in size between ring gage and check, or master, which may affect some of the items.

From the above summary these deductions may be made: The third scheme (Fig. 8) increases the minimum looseness (allowance) by an amount equaling nine times the master gage tolerance (from 2 to $2\frac{1}{4}$ times as much as in the other schemes). It also reduces the net tolerance available in the manufacture of the product nearly three times as much as do the other schemes.

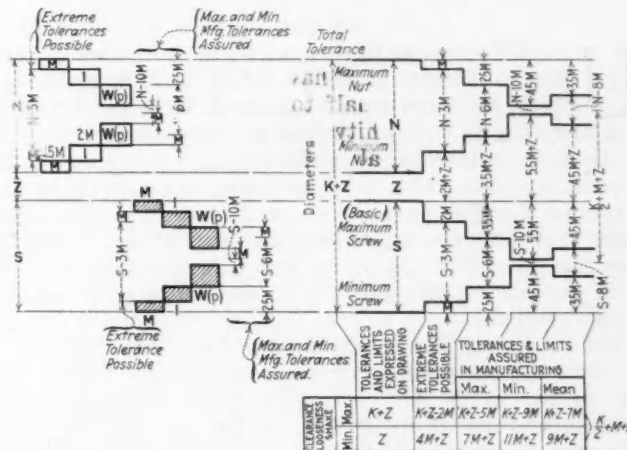


FIG. 8—SCHEME OF THREAD GAGE TOLERANCES USED TO SOME EXTENT IN UNITED STATES

The conclusion one must draw is that the manufacturer loses the tolerance to which he is entitled, this largely increasing the "allowance" which could easily be provided for fully in the specification and without loss to him.

The following concrete example (7-10 threads per inch) offers a good illustration:

Nominal	Apparent Looseness	Net. Aver. Looseness	Net. Nut Tol. Looseness	Net. Se. Tol. Looseness	Gage Tolerances
	Max. Min.	Max. Min.	Max. Min.	Max. Min.	
Total nut tol. f... 0.012	0.006 0.006	0.0280 0.0080	0.0100 0.0080	0.0084 0.0006	Master... 0.0084 } 0.0010
Total allowance... 0.006	0.006 0.006	0.0280 0.0080	0.0100 0.0080	0.0006 0.0006	Inspection... 0.0006 } 0.0020
Total screw tol. f... 0.012	0.006 0.006	0.0280 0.0080	0.0100 0.0080	0.0006 0.0006	Working... 0.0010 }

†As given on specification 30-2-24 U. S. A. Ord. Dept.

‡Net = total tolerance. (†) — (inspection + master gage tolerances)

AUTHOR'S CONCLUSION

In concluding this report it is advisable to say a few words, such as usually go to make up the author's "preface" to his "work." In this instance these remarks have to deal with the nature of this paper. It is not intended as a "paper" in the ordinary sense, therefore it does not go into details that in themselves may be important, but which have not as yet been given more than casual consideration. For this reason it is more or less fragmentary or disconnected. Many important details are passed with but a brief statement or two. To go into them at length would make this report savor of a treatise, which obviously it is not intended to be. There is, on account of the attempts at brevity, the possibility that too concise statements will give impressions not intended.

The purpose has been to refer to the screw-thread details that have received attention both in the discussions and visits of inspection; what few points have been mentioned are merely indicative of the complexity of the screw-thread problem as a whole.

Conventional Propeller Calculations

By F. W. CALDWELL* (Member of the Society)

SEMI-ANNUAL MEETING PAPER

Illustrated with PHOTOGRAPHS AND CHARTS

UNDER war conditions, when there is lack of time for extensive propeller trials and experimental work, we must be able to predict the performance of a propeller as to the horsepower absorbed at given airplane and engine speeds and as to its efficiency under these conditions. It is also necessary to make stress calculations for the purpose of predicting strength, since there is usually no opportunity for a destructive test before the propeller has to be put into production. The main purpose of this paper is to show how such calculations are made.

The first step is to choose a diameter. The chart, Fig. 1, shows the minimum diameter required to maintain the slip as low as 15 per cent. The chart also shows the maximum propeller speed that can be used for any given diameter and horsepower. If the speed of rotation is greater than that shown as the maximum, a smaller diameter must be chosen for the propeller, and there is a consequent loss in efficiency.

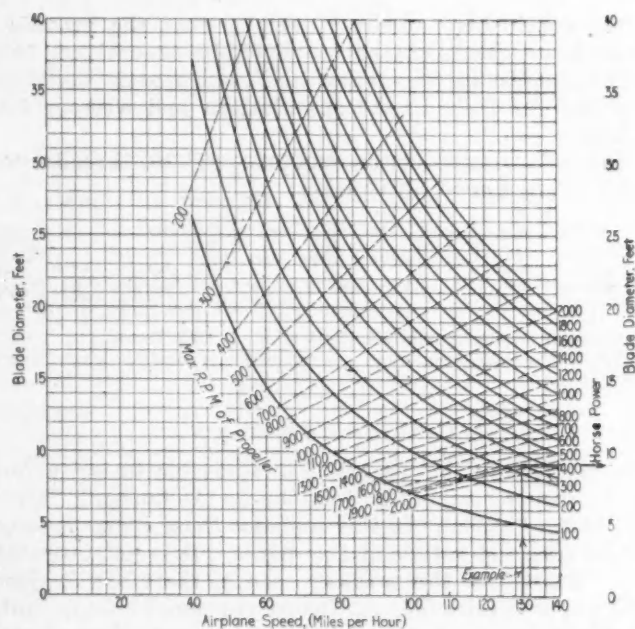


FIG. 1—CHART FOR DETERMINING PROPELLER DIAMETERS FOR 15 PER CENT SLIP. ALSO MAXIMUM PROPELLER SPEEDS FOR GIVEN DIAMETERS AND HORSEPOWERS

The diameters given are minimum diameters for good practice. In general the diameter should be made as large as possible without making the blades so narrow

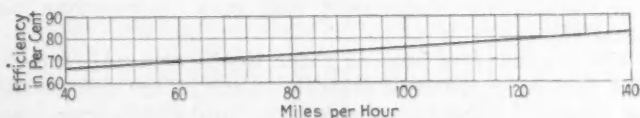


FIG. 1A—PROPELLER EFFICIENCY AT VARIOUS SPEEDS OF FLIGHT USED IN PLOTTING FIG. 1

*Aeronautical mechanical engineer, Airplane Engineering Department, Bureau of Aircraft Production, U. S. Army.

that they will flutter excessively when wood construction is used. Plane designers should bear in mind the necessity of ample propeller diameter in laying out a power plant installation; an otherwise excellent design may be spoiled by limitations of clearance, resulting in a too small propeller diameter.

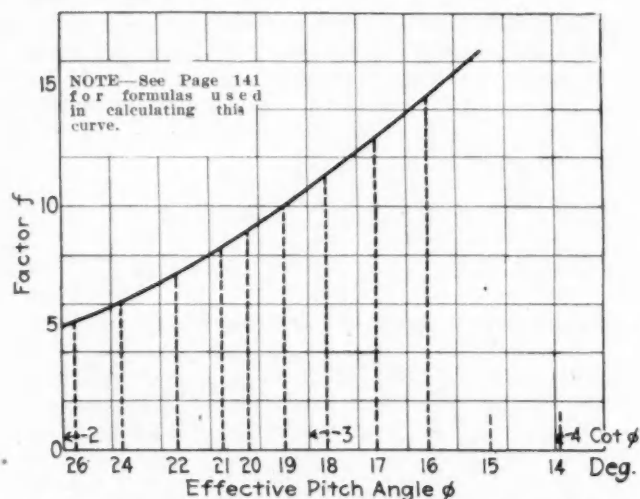


FIG. 1B—VALUES OF FACTOR f FOR EFFECTIVE PROPELLER PITCH ANGLES USED IN DERIVING FIG. 1

After determining the diameter, a blade form must be chosen. The question of the outline of the blade form is an unsettled one, since different blade forms are based on elaborate theories. The difference in efficiency of different blade forms is not great, but the difference in strength is considerable. The form shown in Fig. 2 is a fair one, both as to efficiency and strength.



FIG. 2—EXPERIMENTAL BLADE FORM

The following nomenclature relating to propeller design will be used in this paper:

- ρ = specific weight of air expressed in pounds per cubic foot.
- g = acceleration due to gravity = 32.2 f.p.s.
- K_y = lift coefficient, absolute units.
- K_x = drag (or drift) coefficient, absolute units.
- L = vertical component of force (lift) on aerofoil, pounds.
- D = horizontal component of force (drag) on aerofoil, pounds.
- f = factor for computing work absorbed by propeller.
- C = empirical constant depending on blade form.
- S = area of plane surface, square feet.
- V = velocity of airplane, feet per second.
- P_e = effective pitch of propeller or advance per turn.

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V_1 = velocity (in helical path) of propeller element, feet per second.

v = velocity of slip stream, feet per second.

b = maximum blade width, feet.

b_1 = effective blade width, feet = $0.75b$ for blade form in Fig. 2.

N = engine speed at ground level, rev. per second.

N_1 = engine speed at 20,000 ft. altitude, rev. per second.

D = diameter of propeller, feet.

D_1 = equivalent diameter of propeller, feet = $0.580D$ (for computing work absorbed only).

R = radius of propeller, feet.

T = thrust, pounds.

A = area of propeller disk, square feet.

A_1 = effective area of propeller disk, square feet

$\left(= 0.95\pi \frac{D^2}{4} \right)$ (as corrected for the 5 per cent area considered ineffective due to the radiator).

$\varphi = \tan^{-1} \frac{V}{\pi ND}$ (effective pitch angle).

$\beta = \tan^{-1} \frac{K_y}{K_x}$

θ = blade angle, degrees.

e_1 = theoretical efficiency (Froude Method), per cent

$\left(= \frac{100V}{V + v/2} \right)$

e_2 = aerofoil efficiency, per cent.

e = true efficiency, per cent $(= e_1 \times e_2)$.

speeds above 100 m.p.h. and 3 deg. for plane speeds below 100 m.p.h. The method of checking this would go beyond the scope of this paper, which is intended to cover only fundamentals.

The maximum blade width should be about one-twelfth the diameter for the best practice. This gives an "aspect ratio" of six for each blade.

The usual method of computing blade widths consists in dividing the blade into zones and treating each zone as a separate aerofoil. The power absorbed by each zone is then found from the formula

$$\text{Power for zone} = \frac{\rho}{g} K_x S V_1^3 \quad (2)$$

This may be simplified by taking an average value of K_y for the blade and a weighted mean for the blade width. The power absorbed per blade is then found from the formula:

$$\text{Power} = \frac{\rho}{g} K_y b_1 R V^2 f C \text{ ft.-lb. per sec.} \quad (3)$$

The value of f may be obtained by taking 0.58 of the diameter to represent an equivalent diameter for the whole blade. (This value has been established experimentally for blade form shown in Fig. 2.) Then the value of $V/0.58\pi ND$ is calculated, as is the corresponding angle whose tangent is $v/0.58\pi ND$, and the corresponding value of f is found on the chart, Fig. 3.

The ordinate corresponding to the angle is followed until it crosses the line corresponding to an L/D of the section, and the corresponding value of f is read on the scale at the right or left. An average value of L/D may be assumed with sufficient precision to be twenty, when using this method.

The empirical constant C is dependent on the blade form and must be determined experimentally. For the blade form shown in Fig. 2 it is 1.1, while it varies from 0.85 to 1.2 for different blade forms now in use. The symbol b_1 represents the weighted mean of the blade width. This weighted mean of the blade width is found by determining the mean ordinate of a curve in which the

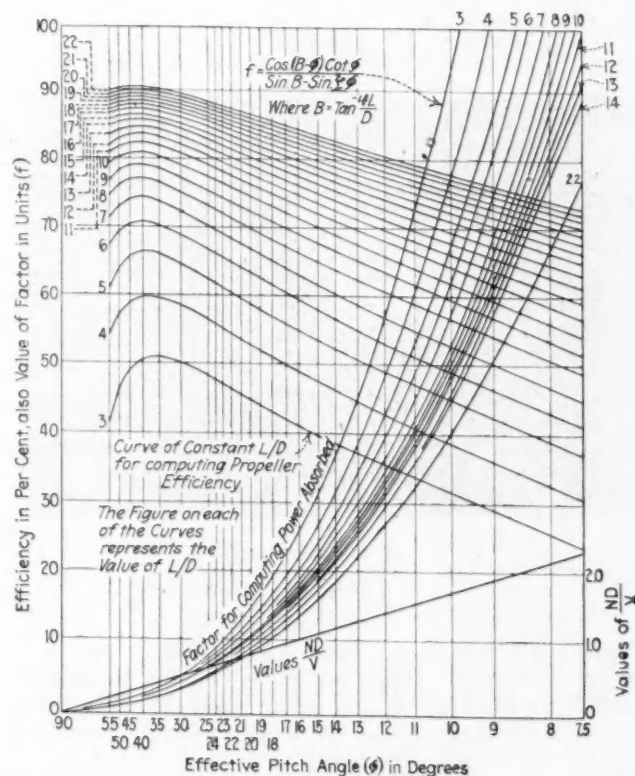
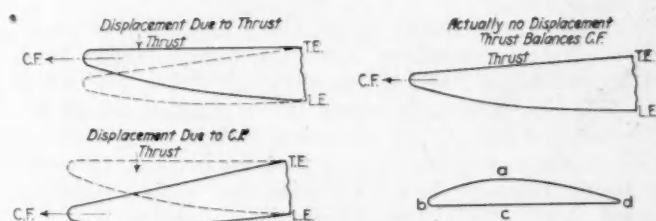


FIG. 3—CURVES USED IN CALCULATING PROPELLER EFFICIENCY, FACTOR f AND $\frac{ND}{V}$

It is necessary to solve for blade width and angle of attack by means of an empirical formula. As a rough rule the angle of attack may be taken as 2 deg. for plane



FIGS. 4, 5, 6—DEFLECTIONS CAUSED BY AXIAL THRUST AND CENTRIFUGAL FORCE. ALSO CROSS-SECTIONAL VIEW OF BLADE

cubes of the radii of the blade sections are laid off as abscissa and the corresponding blade widths are laid off as ordinates. This empirical method gives good results.

The best method of computing propeller efficiency consists in an extension of the water-propeller theories. The theoretical efficiency $V / \left(V + \frac{v}{2} \right)$ is computed. First the thrust is computed and then the slip-stream velocity from the impact formula $T = \frac{\rho}{g} A V v$.

To compute the aerofoil efficiency a representative point along the blade is taken. This will usually be at 75 or 80 per cent of the radius according to the blade shape; for the blade shown in Fig. 2 it is at about 78 per cent.

The product of the theoretical and aerofoil efficiencies gives the actual efficiency very closely. There is a further small correction due to the spiral component of the slip stream.

The angle whose tangent is $V/0.78\pi ND$ is found and a corresponding ordinate ϕ , Fig. 3, is followed until it crosses the efficiency line corresponding to the L/D of the aerofoil section at 0.78 of the radius. This L/D will be about twenty in good design.

PROPELLER MATERIALS

Wood has been the favorite propeller material up to the present. Its success is mainly due to its high tensile strength, light weight and flexibility. Flexibility is an important factor in reducing propeller stresses, as can be seen from Figs. 4 to 6. When the thrust is applied to a wooden propeller, the blade bends and the centrifugal

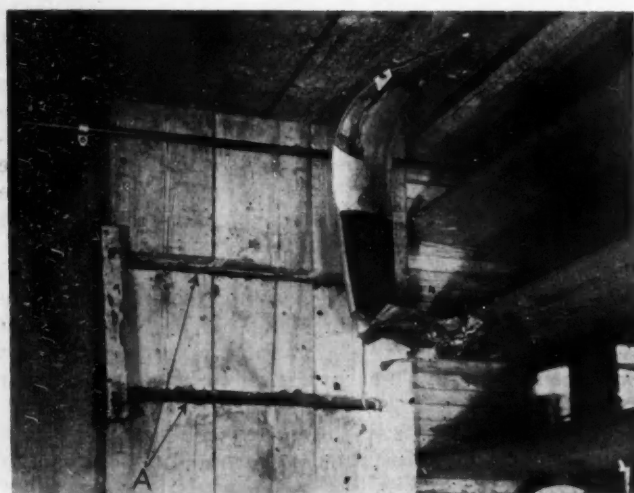


FIG. 7—VIEW AFTER FAILURE OF PROPELLER WITH STEEL BLADE EMBEDDED IN ROOF
A—Observers' slots

force creates a moment tending to restore it to its original position.

The lack of flexibility is evidently one of the weaknesses of steel propellers, since the metal cannot bend and accommodate itself to the different flying attitudes. This difficulty can be overcome for any single flying attitude and air density by offsetting centers of gravity of different sections in such a way that the bending moment due to air pressure is compensated for by the bending moment due to centrifugal force.

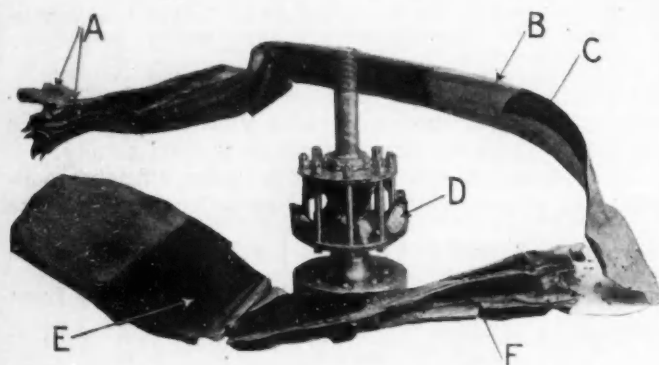


FIG. 8—PARTS OF PROPELLER SHOWN IN FIG. 7
Destroyed after 2 min. at 1330 r.p.m.
A—Hub bolts, sheared 0.065-in. D—Rubber in hub
B—Sheet-steel hub E—Balance hole
C—Rubber coating F—Riveted edges
E—Bare steel

A few photographs of steel-propeller failures are reproduced as Figs. 7 to 14. The propeller shown in Fig. 7 was made of sheet steel riveted along the edges and formed so that the two blades passed each other at the hub, the end of one reinforcing the other. The entire outside of the blades was coated with a layer of hard-rubber compound intended to prevent drumhead vibration. The propeller illustrated in Figs. 9 and 10 was



FIGS. 9 AND 10. UPPER VIEW, STEEL PROPELLER BEFORE WHIRLING TEST. LOWER VIEW SHOWS OPENING (A) AT SEAM AT LEADING EDGE. Failure after 14 min. at 1400 r.p.m. A piece welded inside of blade for balancing flew out of the tip

made of sheet steel, welded by the oxy-acetylene process along the edges and with regular steel hub flanges. None of these propellers was carefully designed, so that the failures occurred after a few minutes' running at comparatively low speeds. All of these propellers were heavier than the wood propellers designed for the same service, and in none of them was the strength of the same order of magnitude as in the wood propellers. We have repeatedly tested wood propellers at speeds up to 2500 r.p.m. without injury.

Among the propeller materials experimented with up to date the metals have shown the least encouraging results. While I do not consider a steel propeller out of

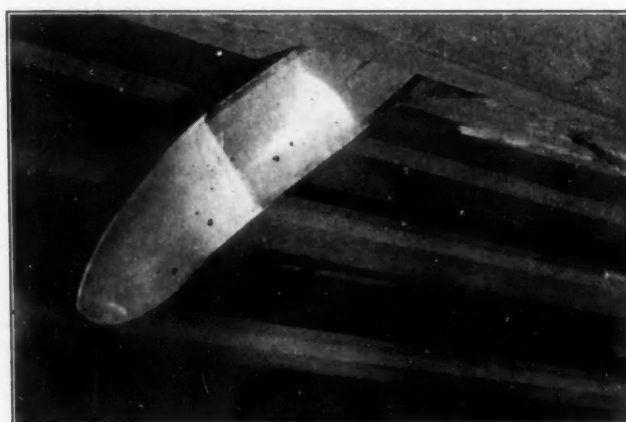


FIG. 11—PROPELLER BLADE EMBEDDED IN ROOF AFTER FAILURE. This propeller consisted of two steel plates riveted to a forged hub. Destroyed after five min. at 1800 r.p.m.

the question, it is certain that the results to date have been discouraging. Steel propellers should be given a thorough ground test before being used in flight on an airplane, since the failures are usually extremely sudden and are disastrous to the plane structure.

Aerofoils

The aerofoil sections used have an important bearing on the propeller efficiency. The characteristics shown in Figs. 15 to 17 are taken from a report issued in 1912 by the National Physical Laboratory of England, and are about as good as any that have been published.

ADJUSTABLE-PITCH PROPELLERS

Almost from the start of air-propeller work, a propeller with adjustable pitch has been considered highly desirable, because it was believed that the efficiency of the propeller could then be maintained constant for different airplane speeds. This is based on the theory that the L/D of the aerofoil section is the controlling factor in the propeller efficiency, a theory which is not borne out in practice.

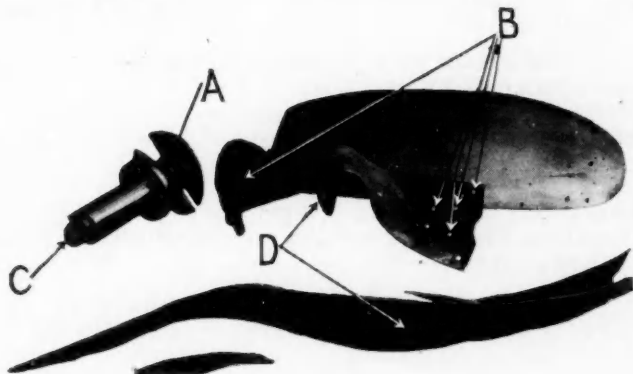


FIG. 12—PARTS OF PROPELLER SHOWN IN FIG. 11
A—Hub uninjured C—Shaft broke
B—All rivets failed in double shear D—Oval rivet sheared

Fig. 18 illustrates the effect of varying plane speed on the apparent angle of attack of a propeller section. This angle is usually chosen as 2 deg. for a flight speed of 130 m.p.h. and the aerofoil will then have a ratio of K_y/K_x of about twenty. In climbing at the rate of 70 m.p.h. the apparent angle of attack will be increased to about 10 deg., and the K_y/K_x ratio will drop to from ten to twelve.

Fig. 19 shows that the increase in true angle of attack is not so great when the slip stream velocity is taken into account; besides this, the theoretical efficiency $V/(V + \frac{v}{2})$ is greatly reduced in climbing and is not increased by an increase in the aerofoil efficiency.

An analysis of an adjustable pitch propeller (appended to this paper) shows no gain in efficiency. There is, however, a net gain in horsepower delivered to the plane, owing to the increase of engine speed in climbing. For the case in question the gain in the rate of climb is 49 per cent. Fig. 20 shows the gain in the rate of climb in a somewhat slower machine.

The keeping up of the engine speed becomes of interest in connection with the development of an engine with torque that is constant at high and low altitudes. From the performance curves shown in Fig. 22 it is apparent that the climbing rate of a plane equipped with such an engine would be greatly improved if the engine speed near the ground were increased and kept the same as the plane climbs.

It is the opinion of the author that, if it becomes desirable, an adjustable pitch propeller of fairly light weight can be built for a smooth-running engine, such as the Liberty twelve, but considerable trouble may be expected with engines that have an inherent vibration.

CONSTANT ENGINE POWER AT ALTITUDE

The author has been told by many engine designers that it would be useless to build an engine to maintain

its power at altitudes, because the propeller efficiency would then be so low that the net gain would be small.

All aeronautical engineers who have made a study of the subject realize that the development of an engine with constant, or nearly constant torque at altitudes up to 20,000 or 30,000 ft., is the one outstanding opportunity for improvement in airplane performance. It is just as easy to design propellers for operation at 20,000 ft. as it is to design them for performance at the ground level, so that the problem is one that must be solved by the engine designers.

The air density at 20,000 ft. is of the order of 50 per cent that at ground level. The density of air is about 0.13 per cent of the density of water. Yet we are using the same means of propulsion in airplanes as is used in boats, and we are obtaining in practice efficiencies as high as 85 per cent, something which cannot be approached in marine practice. Not only is this true, but a propeller designed for use at 20,000 ft. will function without appreciable loss of efficiency near the ground, as may be seen from an analysis in which airplane and propeller performance are worked out in a typical case.

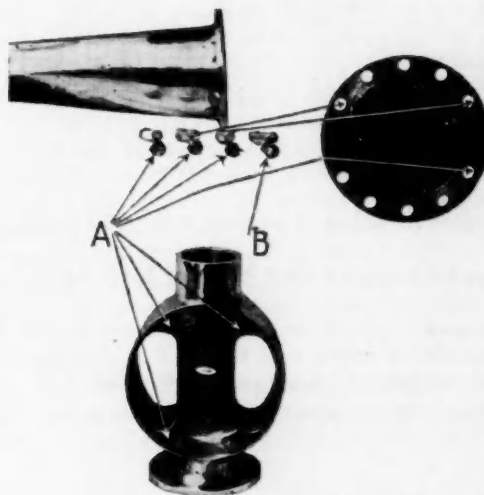


FIG. 13—HUB PARTS OF STEEL PROPELLER AFTER FAILURE, BOTH BLADES UNINJURED
A—Three bolts failed B—One of five bolts uninjured

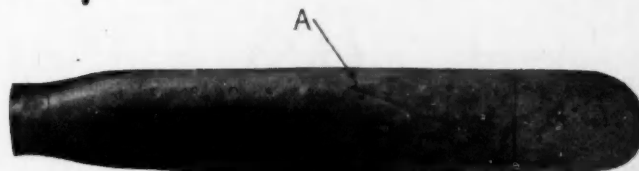


FIG. 14—BLADE OF SIMILAR PROPELLER CRACKED AT A

The words "constant torque" as used in the following analysis mean that the torque is independent of both the engine speed and the altitude. This is the simplest case to discuss, and the discussion applies exactly to this case alone. An engine that satisfies this requirement only in part would, of course, have a performance intermediate between that of the conventional gasoline engine and that of the engine with constant torque. This analysis is applicable equally to a steam turbine and is perhaps of more interest in relation to a steam turbine, owing to the range of speed involved.

In comparing the performance of the airplane and pro-

propeller at 20,000 ft. altitude and at the ground, we will assume:

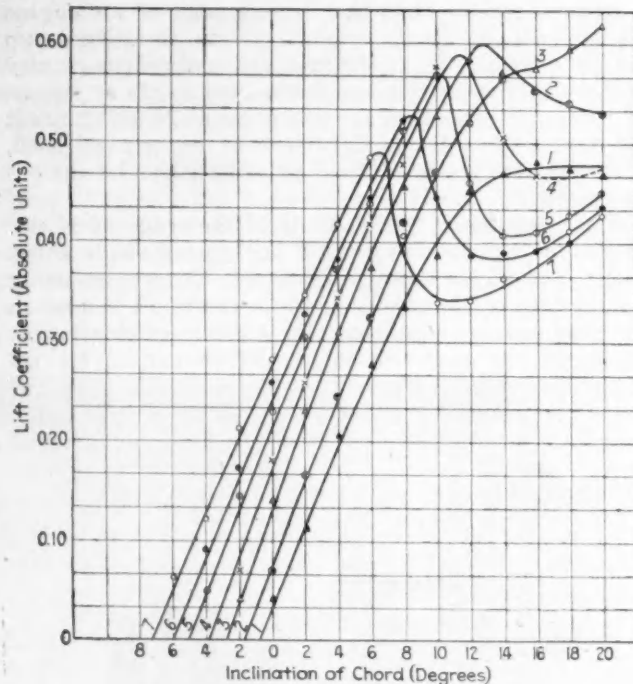


FIG. 15—CHARACTERISTICS OF TYPICAL AEROFOILS FROM N. P. L. REPORT

1. Speed of airplane at ground level, 130 m.p.h., or 191 f.p.s.
2. Output of engine, 356 hp. (at 1400 r.p.m., or 23.3 r.p.s.).
3. Diameter of propeller, 11.5 ft. (disk area 104 sq. ft.).
4. Total lifting surface of plane, $S = 420$ sq. ft.
5. Total weight of loaded plane, $W = 3400$ lb.

The value of K_v in absolute units is computed from the formula

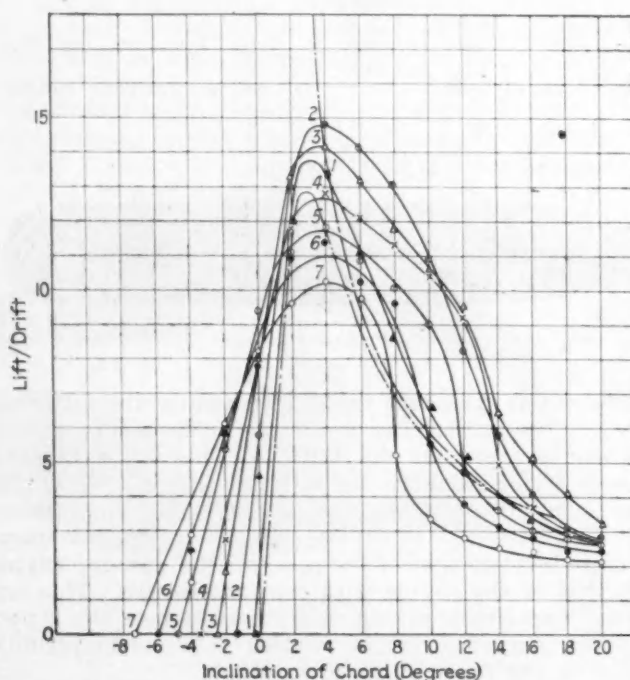


FIG. 16—CHARACTERISTICS OF TYPICAL AEROFOILS FROM N. P. L. REPORT

$$W = \frac{K_v S V^2}{g} \quad (4)$$

$$K_v = \frac{3400}{0.00238 \times 420 \times 191^2} = 0.0955. \quad (5)$$

The corresponding value of K_v/K_x may be taken as 12.3.

The efficiency of the propeller can be computed by a method of trial as follows: First assume an efficiency of 80 per cent. The thrust will then be

$$T = \frac{0.80 \times 356 \times 550}{191} = 820. \quad (6)$$

The slip-stream velocity can be computed from the impact formula (2), and then

$$v = \frac{T}{AV \frac{\rho}{g}} = \frac{820}{104 \times 191 \times 0.00238} = 17.3 \text{ f.p.s.} \quad (7)$$

which means that the slip is 9 per cent.

From the Froude method the theoretical efficiency can be found as follows:

$$e_1 = \frac{V}{V + \frac{v}{2}} = \frac{191}{200} = 0.955. \quad (8)$$

ANALYSIS OF PROPELLER EFFICIENCY

Assuming that the section at 0.75 radius is representative of the propeller as a whole and that the value of L/D at this section is twenty, the efficiency by the aero-

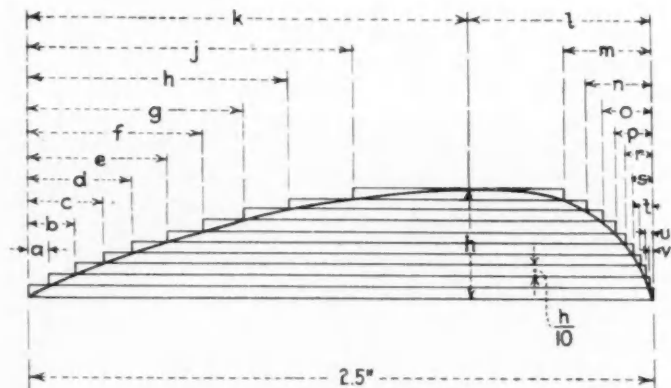


FIG. 17—DIMENSIONS OF TYPICAL AEROFOIL SECTIONS All dimensions in inches (FROM N. P. L. REPORT)

		Plane No.	
a = 0.085	h = 1.060	1	0.063
b = 0.190	j = 1.315	2	0.125
c = 0.305	k = 1.770	3	0.187
d = 0.430	l = 0.730	4	0.250
e = 0.570	m = 0.350	5	0.312
f = 0.715	n = 0.255	6	0.375
g = 0.875		7	0.437
p = 0.140	r = 0.100		
s = 0.070	t = 0.045		
u = 0.025	v = 0.010		
w = 0.010	x = 0.190		

foil method can be obtained from the chart shown in Fig. 3. It is first necessary to find the corresponding value of ND/V , which is

$$\frac{ND}{V} = \frac{1400}{60} \times \frac{0.75 \times 11.5}{200} = 1.01. \quad (9)$$

The aerofoil efficiency is found from Fig. 3 to be 85 per cent. The product of the two efficiencies (see equation 8) is $0.955 \times 0.85 = 81$ per cent. A further correction due to the spiral component of the slip stream will reduce this to 80 per cent. Since the assumption made in computing the slip-stream velocity was correct it need not be recomputed.

The drag of the supporting surfaces at ground level and at a plane velocity of 191 f.p.s. will then be 3400 ÷

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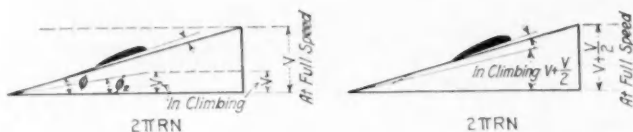
12.3 = 276 pounds. The parasite resistance, that is to say, the propeller thrust minus the wing drag will be $820 - 276 = 544$ pounds.

At 20,000 ft. altitude the air density can be taken as 50 per cent of the density at ground level. As a first approximation assume that the propeller speed will increase in proportion to the cube root of the horsepower (the horsepower being proportional to the engine speed, since constant torque is assumed), and in inverse proportion to the cube root of the density; then in level flight

$$N_1 = 23.3 \sqrt[3]{\frac{N_2}{23.3} \times \frac{1}{0.5}} = 33 \text{ r.p.s.} = 1980 \text{ r.p.m.} \quad (10)$$

Assuming a constant torque the engine will then deliver 503 hp. Assuming 80 per cent propeller efficiency

$$T = \frac{503 \times 0.80 \times 550}{V} = \frac{221,500}{V}$$



FIGS. 18 (LEFT) AND 19 (RIGHT)—EFFECT OF PLANE VELOCITY ON ANGLE OF ATTACK

As a first approximation to determine K_x , assume the velocity to be proportional to the cube root of the horsepower delivered and inversely proportional to the cube root of the air density. Then

$$V = 191 \sqrt[3]{\frac{1}{0.5} \times \frac{503}{356}} = 270 \text{ f.p.s.}$$

$$\text{and } K_y = \frac{3400}{420 \times 0.00119 \times 270^2} = 0.0933.$$

The corresponding K_y/K_x will be 12.3 and K_x will be 0.00759.

Thrust = Total drag =

$$\left(\frac{\rho}{g} K_x S V^2 \right) + \left(\frac{0.5}{1} \times 544 \times \frac{V^2}{191^2} \right)$$

Since $\frac{\rho}{g} = 0.00119$ at 20,000 ft. altitude, we have

$$\frac{221,500}{V} = \left(0.00119 \times 0.00759 \times 420 + \frac{0.5 \times 544}{191^2} \right) V^2$$

Whence

$$V = 270 \text{ f.p.s.} = 184 \text{ m.p.h.}$$

Thus the assumptions as to plane velocity are seen to be correct for constant torque, so that they need not be recomputed.

To find the propeller efficiency under the new condition the thrust is first computed from the formula

$$T = \frac{550 \text{ hp.} \times e}{V} = \frac{550 \times 503 \times 0.80}{270} = 820 \text{ lb.}$$

Again the slip-stream velocity will be

$$v = \frac{T}{AV \frac{\rho}{g}} = \frac{820}{104 \times 270 \times 0.00119} = 24.3 \text{ f.p.s.}$$

The theoretical efficiency will be

$$e = V / \left(V + \frac{v}{2} \right) = \frac{270}{270 + 12} = 0.958.$$

Since ND/V has the same value as before, the true angle of attack will be the same and the aerofoil efficiency will again be equal to 85 per cent.

The product of the two will as before be 81 per cent, and this will be reduced to 80 per cent by the correction for the spiral component of the slip stream.

It is apparent therefore that the efficiency of the propeller is the same for the two flying conditions, provided the engine torque is kept constant.

These results can be deduced without making the calculations as follows: The total lift of the plane may be expressed as

$$W = \frac{\rho}{g} K_y S V^2$$

in which S is constant for all altitudes and W and g are approximately constant. In order to maintain a constant angle of attack K_y must be held constant, then ρV^2 is constant, and V is inversely proportional to the square root of the density. For this condition the total drag of the machine will remain constant and the horsepower required to drive the plane will be equal to the horsepower required at the ground level multiplied by the ratio of the plane velocity at altitude to the plane velocity at the ground.

If the speed of the propeller is kept proportional to the velocity of the plane, its ND/V and its efficiency will be constant. To maintain this speed the engine horsepower must be proportional to the ratio of N^2 multiplied by the inverse ratio of the air densities. But since N^2 is proportional to V^2 it is also inversely proportional to the ratio of air densities. It is consequently necessary to maintain only the engine horsepower proportional to the speed. This is obviously the definition of an engine with constant torque.

The rate of climb to be expected from a plane fitted with a constant-torque engine is shown graphically in Fig. 21. (The calculations for rate of climb are given

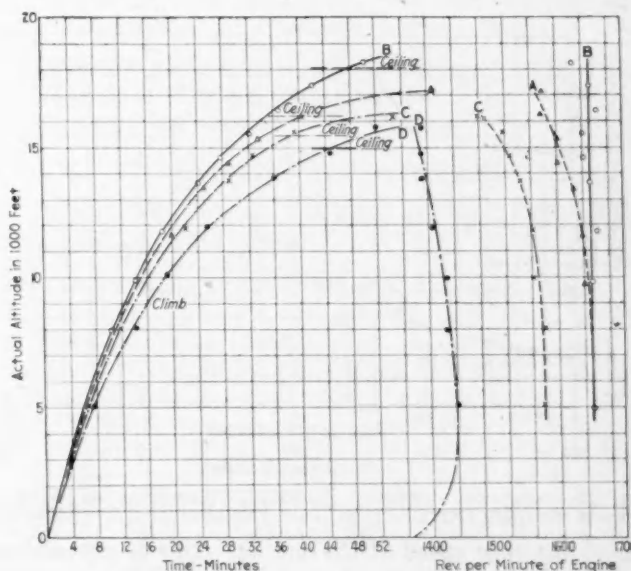


FIG. 20—COMPARISON OF CLIMBING TESTS OF ADJUSTABLE-PITCH PROPELLER

A—Pitch set at ground C—Pitch set to check with D
B—Pitch changed during climb D—Solid wood blade

at the end of the paper.) If an adjustable-pitch propeller were applied to this engine, the propeller speed could be maintained at 1700 r.p.m. during the whole climb. The horsepower available and the time required for climbing, Figs. 22 and 23, indicate such an improvement in the performance that the variable pitch feature becomes highly desirable.

Ceiling

Evidently the ceiling of such a plane is not limited by aerodynamical considerations, but only by the strength of the materials employed. The engine and propeller speed would continue to increase until something let go. The limiting speed of a propeller of this size would probably be about 2500 r.p.m. in thin air with the materials now in use. This would correspond to an air density equal to 31 per cent of that at the ground, and at an altitude of about 34,000 ft. In flying at greater heights it would be necessary to throttle the engine in order not to over-stress the propeller, so that the ceiling would be in the neighborhood of 45,000 feet.

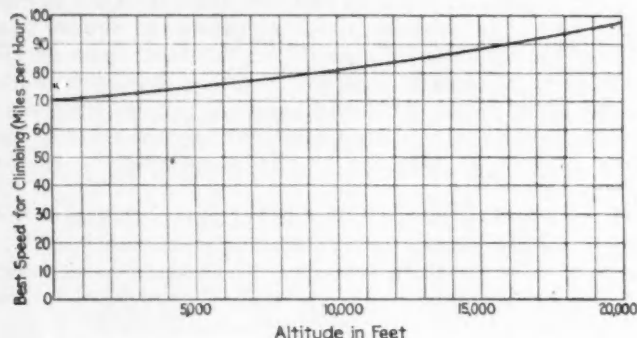


FIG. 21—RATE OF CLIMB OF PLANE FITTED WITH CONSTANT-TORQUE ENGINE

For absolute value of K_y , L/D and wing drag are constant. Power absorbed by drag of wings will then be proportional to speed of plane

In conclusion I wish to emphasize the fact that the variable-pitch propeller is, to a certain extent, limited to special cases. The design of a propeller for an engine with constant torque presents no difficulty, except that the climbing rate near the ground must be reduced, ow-

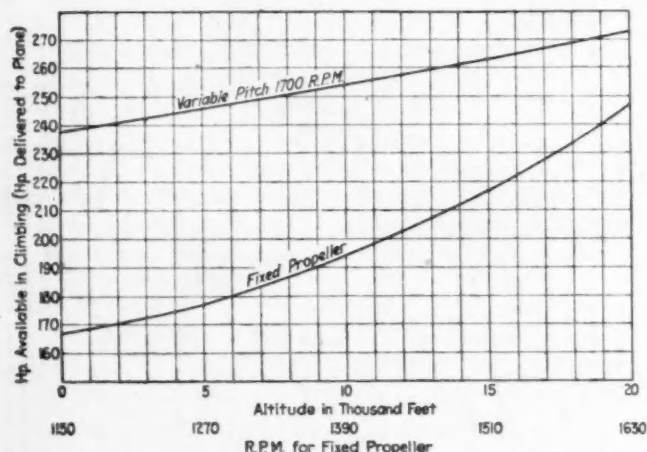


FIG. 22—CLIMBING POWER AT DIFFERENT ALTITUDES (CONSTANT-TORQUE ENGINE) WITH FIXED AND VARIABLE PITCH PROPELLERS

ing to the slow speed of the engine. This is not serious and can be entirely overcome by the use of a variable-pitch propeller. The interesting feature of this development is the great speed to be expected. It is not out of the question to attain speeds of 200 m.p.h. at an altitude of 30,000 feet.

CONVENTIONAL DESIGN COMPARED WITH THAT OF ADJUSTABLE PITCH PROPELLER

The following assumptions will be made in calculating the climbing rate with a fixed blade propeller:

1. Engine output = 170 hp.
2. Propeller diameter = 8 ft. and $A = 50.3$ sq. ft.
3. Radiator area = 3.3 sq. ft. Effective area = $50.3 - 3.3 = 47$.
4. Velocity of airplane = 120 m.p.h. = 176 f.p.s.
5. Engine speed = 1600 r.p.m. = 26.7 r.p.s.

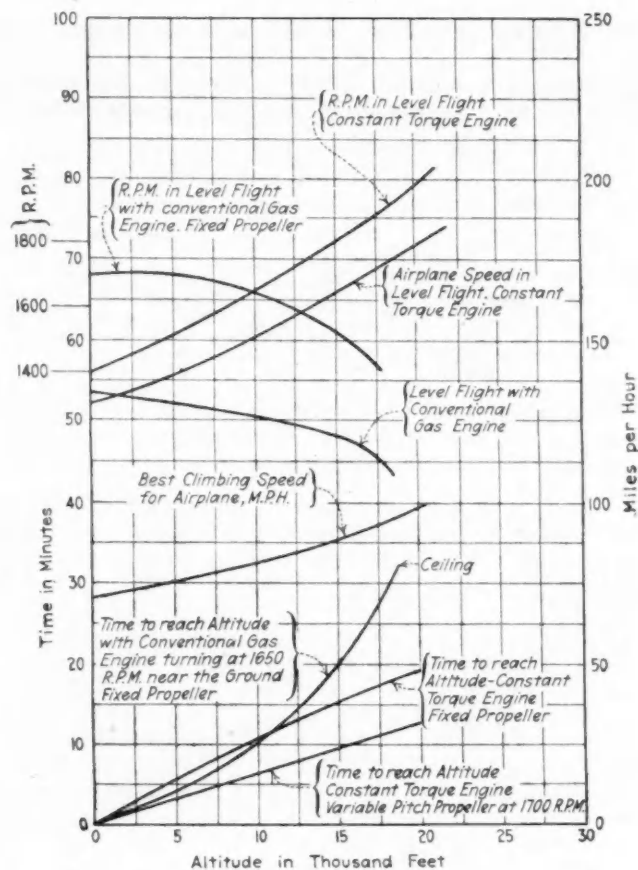


FIG. 23—CLIMBING AND LEVEL FLYING PERFORMANCE OF AIRPLANE WITH CONSTANT TORQUE ENGINE COMPARED WITH THAT OF PLANE WITH CONVENTIONAL GAS ENGINE

The calculations are as follows:

In order to compute the thrust assume a propeller efficiency of 79 per cent.

$$T = \frac{550 \times \text{hp.} \times e}{V} = \frac{550 \times 170 \times 0.79}{176} = 420 \text{ lb.}$$

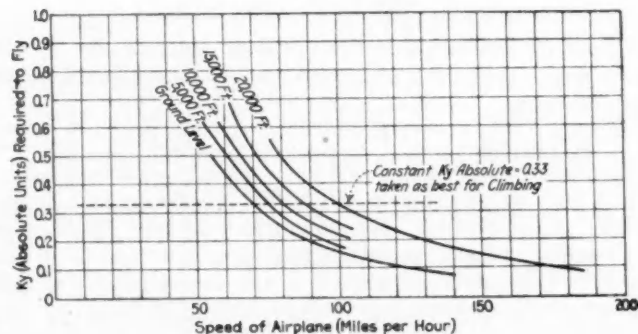


FIG. 24—PROPELLER FOR CONSTANT TORQUE ENGINE. BEST CLIMBING SPEED AT GROUND LEVEL 70 M.P.H.

CONVENTIONAL PROPELLER CALCULATIONS

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$$v = \frac{T}{AV \frac{\rho}{g}} = \frac{420}{47 \times 176 \times 0.00238} = 21.4 \text{ f.p.s.}$$

$$\text{Slip} = \frac{v}{V} = \frac{21.4}{176} = 12.2\%$$

$$e = \frac{V}{V + \frac{v}{2}} = \frac{176}{186.7} = 94.2\%$$

We must now determine the value of $\frac{ND}{V}$ and in doing this it may be assumed that the section at 0.75 radius is representative of the propeller as a whole.

$$\frac{ND}{V} = \frac{0.75 \times 1600 \times 8}{60 \times 186.7} = 0.858.$$

$$\frac{K_y}{K_x} = 20, \quad e_s \text{ (from Fig. 3)} = 0.865.$$

Then $e = 0.942 \times 0.865 = 0.815$, which is reduced to 80 per cent by the spiral component of the slip stream. (The assumption of 79 per cent in computing thrust is nearly enough correct so that it need not be recomputed.)

$$\text{Output} = 0.80 \times 170 = 136 \text{ hp.}$$

$$\text{The effective pitch angle } \varphi = \cot^{-1} \frac{0.75\pi ND}{V + \frac{v}{2}} = 20.3 \text{ deg.}$$

Assume that the true angle of attack from chord is 0.5 deg. and that blade angle (θ) = 20.8 deg.

Propeller Efficiency Under Climbing Conditions

The following calculations apply to propellers on a conventional-type engine:

Assume rate of climb as 65 m.p.h. = 95.3 f.p.s.

The speed in climbing = 1475 r.p.m.

Assume efficiency = 60 per cent.

$$T = \frac{0.60 \times 550 \times 1475 \times 170}{95.3 \times 1600} = 542 \text{ lb.}$$

$$v = \frac{542}{47 \times 95.3 \times 0.00237} = 50.9.$$

$$\text{Slip} = \frac{50.9}{95.3} = 53.4\%.$$

$$e_s = \frac{V}{V + \frac{v}{2}} = \frac{95.3}{120.8} = 79\%.$$

$$\frac{ND}{V} = \frac{0.75 \times 1475 \times 8}{60 \times 120.8} = 1.22.$$

Effective pitch angle $\varphi = 14.5$ deg.

Angle of attack = blade angle — effective pitch angle = 20.8 — 14.5 = 6.3 deg.

$$\frac{K_y}{K_x} = 15.5, \quad e_s \text{ (from Fig. 3)} = 0.79.$$

$e = 0.79 \times 0.79 = 62.3$ per cent, which is reduced by the spiral component of slip stream to 60 per cent.

$$\text{Output} = \frac{0.60 \times 1475 \times 170}{1600} = 94 \text{ hp.}$$

Before going further to determine the climbing rate let us obtain figures similar to the above for an adjustable pitch propeller. For this assume that

Speed = 1970 r.p.m. during climbing.

$$\text{Output} = \frac{1970 \times 170}{1600} = 209 \text{ hp.}$$

Efficiency = 55%.

Air speed = 65 m.p.h. = 95.3 f.p.s.

$$T = \frac{0.55 \times 550 \times 209}{95.3} = 655 \text{ lb.}$$

$$v = \frac{665}{47 \times 95.3 \times 0.00238} = 62.5 \text{ f.p.s.}$$

$$\text{Slip} = \frac{62.5}{95.3} = 65.5\%.$$

$$e_s = \frac{V}{V + \frac{v}{2}} = \frac{95.3}{126.7} = 75.3\%.$$

$$\frac{K_y}{K_x} = 20, \quad e_s \text{ (from Fig. 3)} = 0.85.$$

$$\frac{ND}{V} = \frac{0.75 \times 1970 \times 8}{60 \times 126.7} = 1.55.$$

$e = 0.86 \times 0.753 = 64.7$ per cent, which is reduced by spiral component of slip stream to be 59 per cent.

$$\text{Output} = 0.59 \times 209 = 123 \text{ hp.}$$

Comparison of Climbing Rates

Assume

Weight = 1400 lb.

Power required to fly at 65 m.p.h. = 35 hp.

With the Fixed Pitch Propeller

Excess power for climbing = 94 — 35 = 59 hp.

$$\text{Rate of climb is } \frac{59 \times 33,000}{1400} = 1390 \text{ f.p.m.}$$

With Adjustable Pitch Propeller

Excess power for climbing, 123 — 35 = 88 hp.

$$\text{Rate of climb, } \frac{88 \times 33,000}{1400} = 2070 \text{ f.p.m.}$$

Gain in rate of climb = 49%.

The increasing speed of engine during climbing will have the tendency to overspeed the powerplant while climbing and to allow the speed to become normal in level flight. The desirability of this feature is still open to question.

Calculations of Efficiency During Climbing

The following calculations apply to an adjustable pitch propeller on a constant torque engine:

(1) At ground level:

$$\frac{\rho}{g} = 0.00238 \quad A = 104 \text{ sq. ft.}$$

$V = 103$ f.p.s. Engine output, 445 hp. at 1700 r.p.m.
Assume efficiency = 55%.

$$T = \frac{0.55 \times 445 \times 550}{103} = 1310 \text{ lb.}$$

$$v = \frac{1310}{0.00238 \times 104 \times 103} = 51.4 \text{ f.p.s.}$$

$$e_s = \frac{V}{V + \frac{v}{2}} = \frac{103}{128.7} = 80\%.$$

$$\frac{ND}{V} = \frac{1700 \times 11.5 \times 0.75}{60 \times 138.6} = 1.764.$$

$$\frac{K_y}{K_x} = 16, \quad e_s \text{ (from Fig. 3)} = 0.720.$$

$e = 0.720 \times 0.80 = 0.576$, which is reduced by the spiral component of the slip stream to be 53 per cent.

$$\text{Output (useful)} = 445 \times 0.53 = 236 \text{ hp.}$$

(2) At 10,000 ft. altitude:

$$\frac{\rho}{g} = 0.00178 \quad A = 104 \text{ sq. ft.}$$

$$V = 118.7 \text{ f.p.s. (See Fig. 21.)}$$

Output = 445 hp. at 1700 r.p.m.

Assume efficiency = 56%.

$$T = \frac{0.56 \times 445 \times 550}{118.7} = 1155 \text{ lb.}$$

$$v = \frac{1155}{0.00178 \times 104.7 \times 118.7} = 52.5 \text{ f.p.s.}$$

$$e_1 = \frac{V}{V + \frac{v}{2}} = \frac{118.7}{145} = 0.819.$$

$$\frac{ND}{V} = \frac{1700 \times 11.5 \times 0.75}{60 \times 153.9} = 1.59.$$

$$\frac{K_y}{K_x} = 16 \cdot e_2 \text{ (from Fig. 3) } = 0.74.$$

$e_1 = 0.819 \times 0.74 = 60.5$ per cent, which is reduced by spiral component of slip stream to be 57 per cent.

Output (useful) = $445 \times 0.57 = 254$ hp.

(3) At 20,000 ft. altitude:

$$\frac{\rho}{g} = 0.00119. \quad A = 104 \text{ sq. ft.}$$

$$V = 144 \text{ f.p.s. (See Fig. 21.)}$$

Output = 445 hp. at 1700 r.p.m.

Assume efficiency = 62%.

$$T = \frac{0.62 \times 445 \times 550}{144} = 1055 \text{ lb.}$$

$$v = \frac{1055}{0.00119 \times 104 \times 144} = 61.7 \text{ f.p.s.}$$

$$e_1 = \frac{V}{V + \frac{v}{2}} = \frac{144}{175} = 0.823.$$

$$\frac{ND}{V} = \frac{1700 \times 11.5 \times 0.75}{185 \times 60} = 1.32.$$

$$\frac{K_y}{K_x} = 16 \cdot e_2 \text{ (from Fig. 3) } = 0.79$$

$e = 0.79 \times 0.823 = 65$ per cent, which is reduced by spiral component of slip stream to 62 per cent.

Output (useful) = $445 \times 0.62 = 276$ hp.

Engine with Constant Torque

The values of K_y necessary to maintain flight at different speeds and altitudes can be calculated from the formula:

$$\frac{\rho}{g} K_y S V^2 = 3400$$

in which S equals 420, so that

$$K_y = \frac{8.1}{\frac{\rho}{g} V^2}$$

These values of K_y are given in the following table:

M.P.H.	40	60	80	100	120	140
F.P.S.	58.7	88.0	117.3	146.7	176.0	205.3
Altitude, Ft.	$\frac{\rho}{g}$					
0	0.00237	0.996	0.441	0.249	0.159	0.110
2,000	0.00225	0.465	0.262	0.167	0.107	0.081
4,000	0.00213	0.491	0.277	0.177	0.113	0.086
6,000	0.00201	0.521	0.293	0.187	0.119	0.091
8,000	0.00189	0.554	0.312	0.199	0.126	0.096
10,000	0.00178	0.588	0.331	0.212	0.133	0.101
12,000	0.00167	0.627	0.353	0.226	0.140	0.106
14,000	0.00155	0.671	0.380	0.243	0.148	0.111
16,000	0.00143	0.720	0.414	0.263	0.156	0.116
18,000	0.00131	0.774	0.449	0.288	0.165	0.121
20,000	0.00119	0.833	0.495	0.317	0.175	0.126

The rate of climb can be obtained by computing the thrust and the wing drag. The difference between thrust and wing drag at maximum speed will be the parasite resistance. The parasite resistance for other speeds can be figured from the ratio of the cube of the velocities, and the wing drag by the usual method. From the sum of these two the power required to fly can be computed and when this is subtracted from the useful power delivered the excess horsepower and consequently the rate of climb can be figured.

For 130 m.p.h. (191.7 f.p.s.),
assume $K_y = 0.0935$. $\frac{K_y}{K_x} = 12.3$.

$$\text{Wing drag} = W \div \frac{K_y}{K_x} = \frac{3400}{12.3} = 276 \text{ lb.}$$

$$\text{Power delivered to plane} = 0.80 \times 356 = 285 \text{ hp.}$$

$$\text{Wing drag hp.} = 191.7 \times \frac{276}{550} = 96.$$

Assuming engine speed of 1150 r.p.m. at the ground, in climbing at 70 m.p.h. the engine will give 293 hp. Assuming 58 per cent propeller efficiency, the thrust will be

$$T = \frac{0.58 \times 293 \times 550}{103} = 911 \text{ lb.}$$

$$v = \frac{911}{0.00237 \times 104 \times 103} = 35.8 \text{ f.p.s.}$$

$$e_1 = \frac{V}{V + \frac{v}{2}} = \frac{103}{120.9} = 0.853.$$

SUMMATION OF RESULTS OF CALCULATIONS FOR ENGINE WITH CONSTANT TORQUE

Altitude	0	5,000	10,000	15,000	20,000
Best climbing speed: M.P.H.	70	75	81	88	98
F.P.S.	102.7	110.0	118.8	129.1	143.7
Wing drag, hp. = $\frac{3400}{K_y/K_x} \times \frac{V}{550}$	44.5	47.5	51.4	55.8	62.3
Parasite resistance, hp. = $189 \rho / V^{1/1913}$	30.0	31.7	34.3	36.5	40.6
Total hp. required to fly	74.5	79.2	85.7	92.3	102.9
Hp. available with conventional type propeller	167.2	178	195	217	247
Excess hp.	92.7	98.8	109.3	124.7	144.1
Rate of climb, f.p.m.	900	960	1062	1210	1400
Time to reach altitude, minutes	5 2/3	5 1/4	10 2/3	15	19
Hp. available with adjustable pitch propeller at 1700 r.p.m.	236	246	254	263	276
Excess hp.	163.5	166.8	168.3	170.7	173.1
Rate of climb, f.p.m.	1610	1615	1635	1660	1650
Time to reach altitude, minutes	3 1/6	3 1/6	6 1/3	9 1/2	12 1/2

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$$\frac{ND}{V} = \frac{1150 \times 11.5 \times 0.75}{60 \times 120.9} = 1.38.$$

Assuming a $\frac{K_y}{K_x}$ of 13 for this condition, the aerofoil efficiency will be 67 per cent, and
 $e = 0.67 \times 0.853 = 57\%$.

CALCULATIONS FOR PROPELLER CHART

The chart shown in Fig. 1 was prepared in part from the curves given in Figs. 1-A and 1-B.

The data used in plotting all these curves were obtained as follows:

V = Velocity of plane, ft. per sec.

v = Velocity of slip stream, ft. per. sec. = $0.15V$.

D = Diameter of propeller, ft.

D_1 = Equivalent propeller diameter = $0.580D$.

The following calculations are used only to compute the power absorbed:

$b_{\frac{\pi}{2}}$ = Max. blade width, ft.

b_1 = Weighted average blade width = $0.75b_{\frac{\pi}{2}}$.

ρ/g = Air density = 0.00237.

e = Efficiency of propeller.

$$T = \text{Thrust} = \frac{Hp. \times e \times 550}{V}$$

$$\rho/g \times K_y = 6.3 \times 10^{-4}.$$

$$\frac{K_y}{K_x} = 20 \text{ (assumed).}$$

f = Factor for power absorbed by blade.

$$= \frac{\cos(\beta - \varphi) \cot \varphi}{\sin \beta \sin^2 \varphi}$$

$$\tan \beta = \frac{K_y}{K_x}$$

Aspect ratio assumed as 6. Number of blades = 2.

Factor for blade shape, $C = 0.95$.

Note that the blade form used is not same as Fig. 2.

$$\text{Effective disk area (A)} = (D^2 - 0.05D^2) \frac{\pi}{4} = 0.95 \frac{\pi}{4} D^2.$$

$$v = \frac{T}{\rho/g \times A \times V}$$

$$0.95 \times \frac{\pi}{4} \times D^2 \times V \times v \times \rho/g = \frac{Hp. \times e \times 550}{V}$$

$$D = 1437.5 \sqrt{\frac{Hp. e}{V^3}}$$

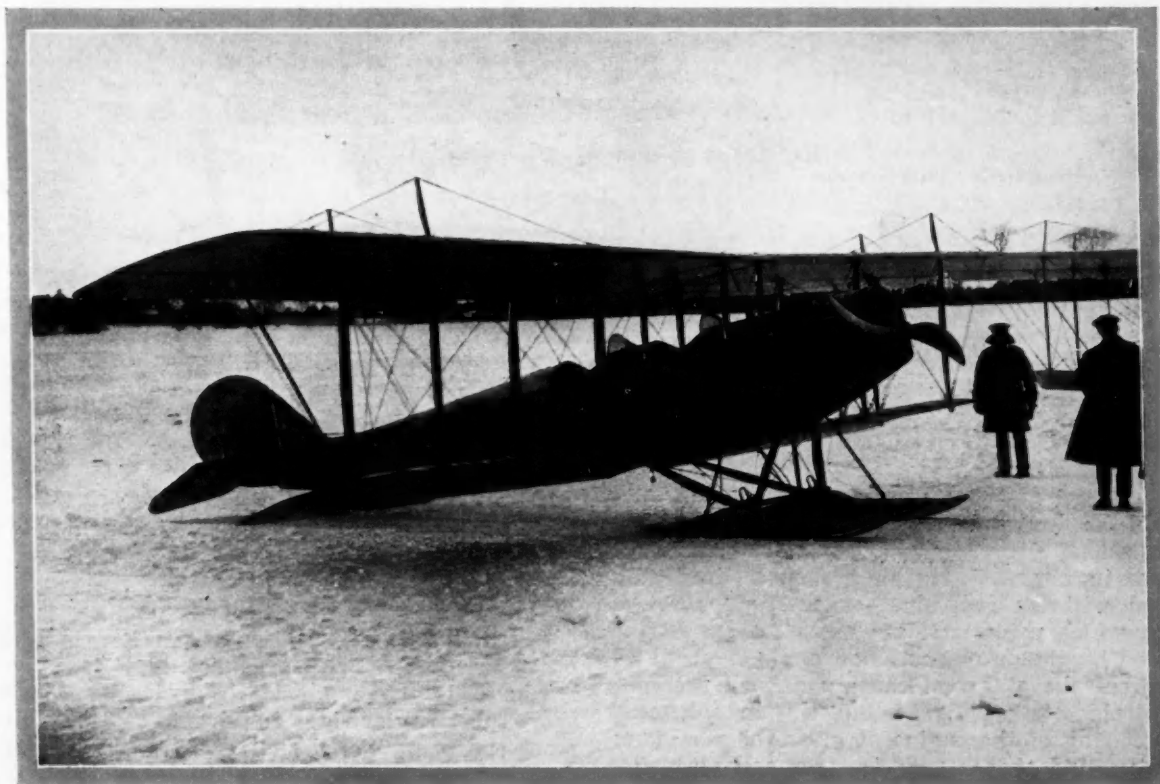
The following are used only in the derivation of the chart in Fig. 1:

$$C \times 2 \times \frac{\rho}{g} K_y \times b_1 \times R \times V^3 \times f = Hp. \times 550.$$

$$f = \frac{8800 Hp.}{\rho/g K_y \times D^2 V^3 \times C}$$

$$\cot \varphi = \frac{2\pi R N}{V} \text{ (At any radius R).}$$

$$\frac{ND_1}{V} = \frac{\cot \varphi}{\pi} \quad \text{R.p.m.} = \frac{32.9 V \cot \varphi}{D}$$



AIRPLANE, BUILT IN A CANADIAN FACTORY, FITTED WITH RUNNERS FOR USE ON SNOW

Exhaust Headers and Mufflers for Airplane Engines

By ARCHIBALD BLACK* (*Member of the Society*)

SEMI-ANNUAL MEETING PAPER

IN the following notes no attempt has been made to treat exhaustively the design of exhaust headers or mufflers for airplane engines. The paper is intended chiefly to record some data collected on this subject during the past several years.

The exhaust header's primary purpose is to carry exhaust gases away from the engine, but it may be designed to perform certain other functions to advantage. The most important of these is probably the muffling of the sound of the exhaust, which may be brought about by a properly designed header.

In present practice the exhaust is taken care of in several radically different ways, the devices of the different designers ranging from the open port to the elaborate combined header and muffler as now used on some machines.

TYPICAL CONSTRUCTIONS

Fig. 1 shows the open port, which was very popular on the old engines and was used until recently on the Curtiss 90-hp. JN-4 type engine. In that type no effort was made to carry away the gases or to muffle the noise.



FIG. 1—EARLY INSTALLATION OF CURTISS 90 HP. ENGINE SHOWING THE OPEN EXHAUST PORTS

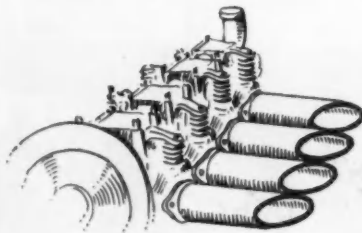


FIG. 2—INDIVIDUAL EXHAUST PIPES ON HALL-SCOTT ENGINE

Fig. 2 shows the "stub tube" method, used on some Hall-Scott installations and recommended by the Hall-Scott Company for its engines. In this system each cylinder is furnished with an individual exhaust pipe several inches long, which points straight out from the engine and is cut off on the outer end at an angle of about 45 deg. to prevent the air from causing a back pressure when the airplane is moving. While this arrangement assists the cooling of the engine, it gives the maximum of noise and would appear therefore not entirely satisfactory for either military or sporting machines.

Fig. 3 illustrates what has come to be known by many

as the German system on account of its origin on machines of that nation and apparent popularity among German designers. Tubes extend upward from each cylinder into the common header, which is of streamline section and points upward and slightly to the rear, directing the gases over the top of the upper wing. This reduces the noise heard from below and appears to be a very practicable, although not final, solution of the problem.

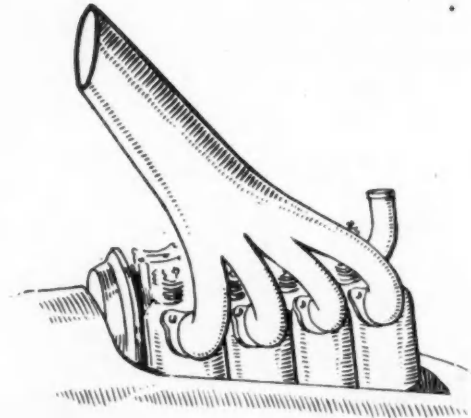


FIG. 3—GERMAN TYPE OF EXHAUST HEADER AS USED ON MERCEDES AND OTHER ENGINES

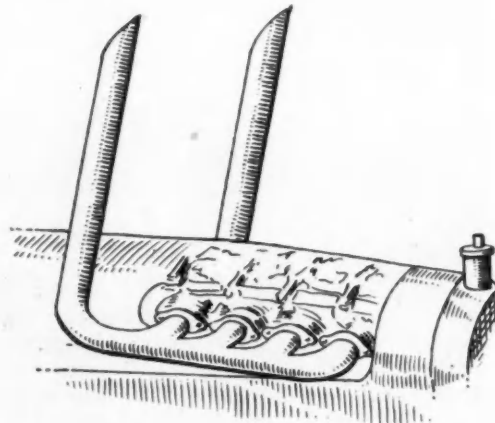


FIG. 4—GERMAN TYPE OF HEADER ON CURTISS JN-4

Fig. 4 shows an adaptation of the German system used on recent Curtiss JN-4 machines.

Fig. 5 is the original L-W-F header consisting of short tubes run at right angles into a manifold tube, the front end being closed and the rear end connected by a flexible metallic tube to a straight pipe leading along the body to the rear of the pilot, the end being open to permit the escape of the gases. This design was discarded, after the first few machines, in favor of the method illus-

*Chief Engineer, L-W-F Engineering Company, Inc.

EXHAUST HEADERS AND MUFFLERS FOR AIRPLANE ENGINES

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trated in Fig. 6. Here the individual pipes sweep backward on a liberal radius into the manifold, and the flexible pipe is replaced by a smooth steel tube, which is perfectly straight on some models, depending upon the engine used.

Figs. 7 show the new header and muffler as fitted to some of the Curtiss machines. This header connects to

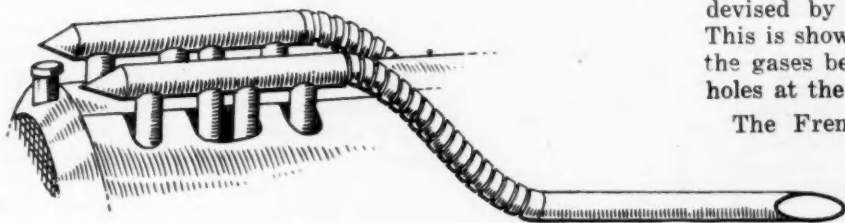


FIG. 5—ORIGINAL L-W-F HEADER

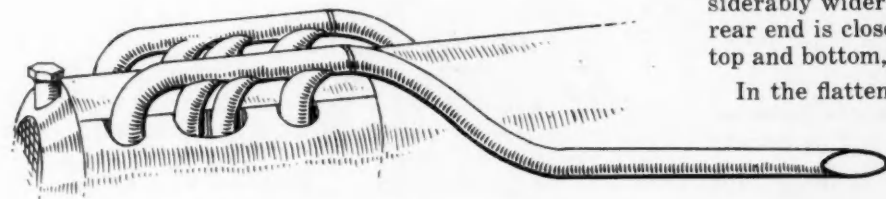


FIG. 6—LATER MODEL OF L-W-F HEADER

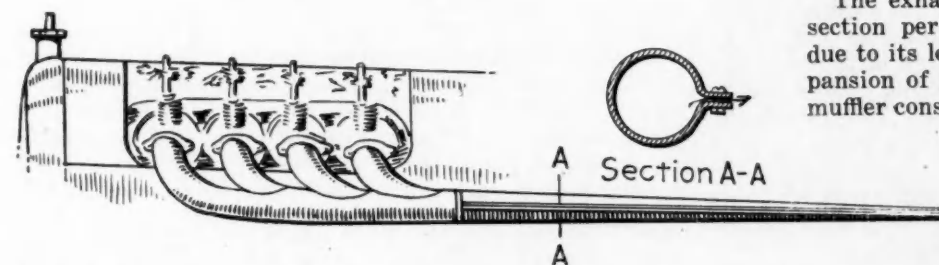


FIG. 7—CURTISS TYPE MUFFLER ON R-4 MACHINE AND FIG. 8—SECTION A-A SHOWING THE SLOT

a tube, which extends backwards along the body and terminates in a muffler. The muffler is simple and most ingeniously designed; the exhaust tube is slotted and the metal bent outward to form two parallel flanges separated by spacers as shown in Fig. 8. The author had no opportunity to examine the earlier of these mufflers, as installed on R-4 machines, but the following description was reported to him by the field department of the L-W-F Company. The slot was about 5 to 6 ft. long and about $\frac{1}{8}$ in. wide, starting near the engine and running back to the end of the tube, which tapered from the full section of the engine to a comparatively small section, open to the atmosphere, at the rear end.

One of the Curtiss engineers stated that no material loss of power was found to be caused by the use of this muffler. It has been generally conceded, by those who have heard the engine running with the muffler in place that the noise of the exhaust is reduced to much less than its former volume. This bears out the author's experience with similar devices installed on several engines; these are referred to later in this paper.

At a later date the author made an examination of a

similar muffler installed on one of the Curtiss Company's JN machines equipped with Hispano-Suiza engines. This muffler had no large opening in the rear, the slot extending around the end instead. The slot was about 4 ft. long and $\frac{1}{16}$ in. wide, approximating an allowance of about 0.04 sq. in. of slot per b.hp.—an extremely small area.

Fig. 9 shows a method of muffling the exhaust gases devised by Deperdussin, the French constructor (c). This is shown as applied to an engine of the rotary type, the gases being carried through internal ducts to outlet holes at the rear of the body.

The French "spad," Fig. 10, is furnished with exhaust headers extending backward for about 6 ft. from the engine. These headers are 3 in. inside diameter and terminate in a muffler consisting of a tube, round at the end, fastened to the exhaust header, which is flattened toward the rear, where it ends up about $\frac{1}{2}$ in. in height and is considerably wider than the diameter of the front end. This rear end is closed except for four slots similar to those in top and bottom, described below.

In the flattened part, both on top and bottom side, are a number of slots 1 in. long and $\frac{1}{8}$ in. wide, with round ends. Each muffler has 78 such slots, giving a total area of 0.1267 sq. in. per brake horsepower for two such mufflers, based on 150 b.hp. (the rating of the Hispano-Suiza).

The exhaust pipe allows 0.08836 sq. in. cross-section per b.hp. and has considerable capacity due to its length. This capacity, by allowing expansion of gases, should assist the action of the muffler considerably. The above figures are exact measurements.

Fig. 11 shows the manifold used on later "Mercedes" engines. No detailed information on the construction of this manifold has been obtained by the author.

Fig. 12 illustrates a manifold used on Farman and Breguet machines, which appears to be similar

in design to the "Mercedes" referred to above. The author has examined the Breguet manifold but could not obtain as much detailed information as desired. The inside apparently contains no baffle plates and the pipes

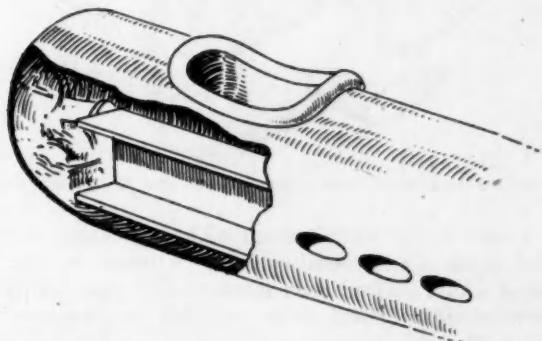


FIG. 9—DEPERDUSSIN EXHAUST DUCT

from the cylinders enter tangentially to the body. This construction is evidently designed to reduce the velocity of the gases by causing them to expend their energy in eddy currents in the main chamber.

Fig. 13 is a scale drawing of one of the mufflers tested under direction of Prof. W. T. Fishleigh (e) at the University of Michigan a few years ago. Five mufflers of different designs were experimented with and detail reports were published in *Horseless Age*, May, 1915, issue (d). Of those tested, the muffler shown here was rated as the best in muffling ability and low back pressure, and one of the two best in low horsepower losses. This

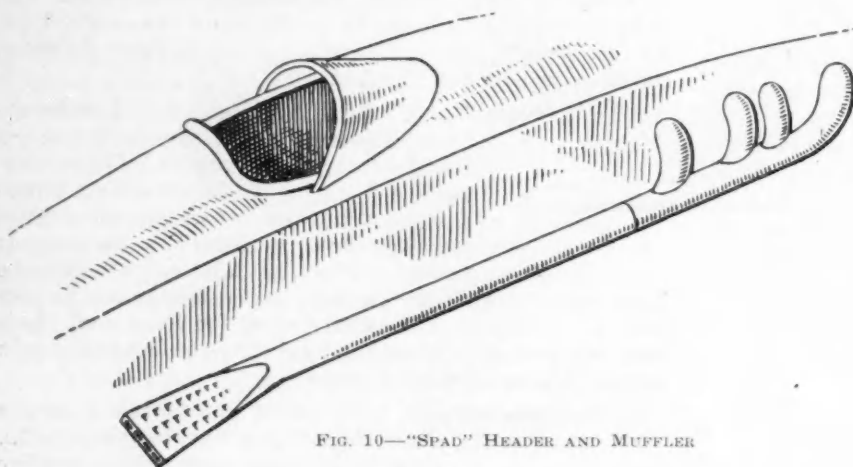


FIG. 10—"SPAD" HEADER AND MUFFLER

muffler was also the lightest of the five. Although the most efficient of those tested, it showed a b.h.p. loss of 3.6 per cent with engine delivering 38 b.h.p. This device weighed 14.5 lb. and had a capacity of 847 cubic inches.

Fig. 14 is a scale drawing of one of the mufflers tested under the direction of Profs. H. Diederichs and G. B. Upton of Cornell University (a). Details of the tests are given in the Second Annual Report of the United States Advisory Committee for Aeronautics. The results of tests of the particular muffler shown are given in part in Table I. It will be noted that this muffler caused a loss in horsepower of only 1.5 per cent, from which it would appear that it is efficient enough to justify its application in many cases to airplane engines.

Another type of muffler has been experimented with recently by J. L. Cato of the L-W-F Company. This muffler was somewhat similar to the "Spad" device, but

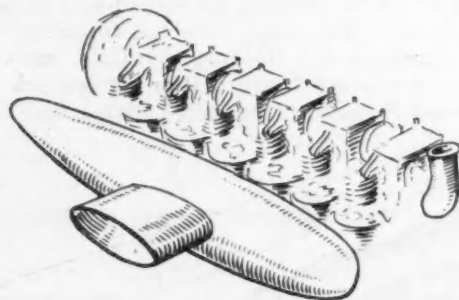


FIG. 11—EXHAUST MANIFOLD USED ON RECENT MERCEDES ENGINES

it had a slot in the end instead of being closed and was provided with small semi-circular louvers in the side instead of slots. The areas allowed and the results obtained were in general very similar to those of the "Spad" muffler.

NOTES ON MANIFOLD DESIGN

The two main considerations in the design of exhaust headers and mufflers are the elimination of back pressure and the reduction of noise, which two requirements

are, unfortunately, generally incompatible in practice. It is a simple matter to design headers fulfilling one, but not both conditions. The combining of both calls for a careful study and proportioning of even the smallest features of the device.

Research work of the manufacturers of blowers and air moving machinery is of considerable help in this work. F. L. Busey conducted a series of experiments (g) to determine the effects of bends in air ducts. The curves published by him, combined here in Fig. 15, show that, when the radii are small, bends in square pipe offer more resistance than bends of the same radius in round pipe. The reverse is the case

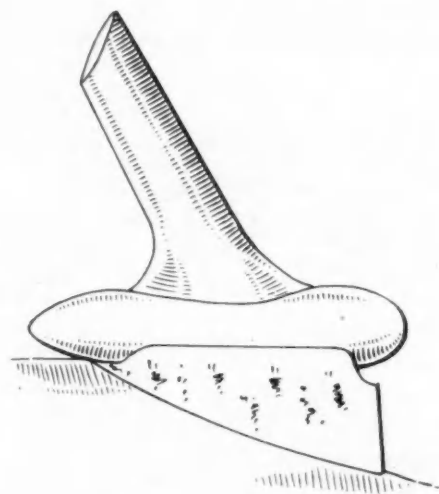


FIG. 12—MANIFOLD USED IN ENGINES IN RECENT FARMAN AND BREGUET MACHINES

when radii are large. This change of condition takes place when the radii equal about 0.6 times the diameter or side.

The American Blower Company (b) also has made some tests and publishes among its data sheets one giving the resistance of bends in terms of equivalent length of straight pipe.

It will be noted from Fig. 15 that the resistance of the bend is lowest when the radius is equal to $2\frac{1}{2}$ times the diameter; that nothing is gained by making it greater than this; and that this resistance rises to a prohibitive value if the radius is less than 1 to $1\frac{1}{2}$ times the diameter of the pipe. While this curve is based on tests which were made at lower speeds than are probably encountered in an exhaust manifold, Willis H. Carrier (f) concluded, from results of experiments, that losses in elbows through which air is flowing depend on the radius of curvature of the elbow and not on its size or on the velocity of the air.

It would therefore appear safe to assume that the curve given applies equally to the high velocity of flow in an exhaust manifold. Bends of high resistance in manifolds, if near the engine, will sometimes show up by becoming red hot. Sudden changes in the area of the exhaust pipe should be avoided. The areas of the individual exhaust pipes are usually fixed by the necessity of making their ends conform to the ports in the cylinders in order to obtain this condition. When this is not the governing condition, personal experience with

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different installations of headers and some investigation of the areas of exhaust ports on several successful aircraft engines suggest that 0.14 to 0.16 sq. in. of area per b.hp. of cylinder is a liberal allowance. The exhaust ports of the Hall-Scott A-5-A, Thomas 8, Sturtevant 5A and Liberty, some of the engines studied, averaged 0.1668 sq. in. per b.hp. As large a radius as possible should be used in these tubes which enter the manifold. The latter should be of large sectional area at the outlet end, tapering down to equal the area of the individual pipe at the other end. Where a tube extension is used it should be as straight as possible and of the same sectional area as the manifold end.

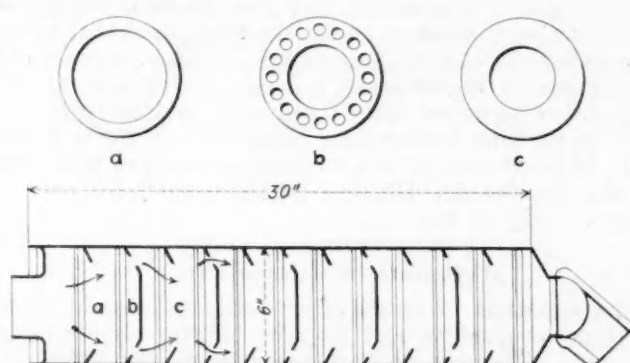


FIG. 13—MUFFLER TESTED AT UNIVERSITY OF MICHIGAN

The proper area for the large end of the manifold is best determined by experiment with several different sizes, noting the effect upon the horsepower of the engine while running on the dynamometer. This best size will probably vary with different engines, being governed, no doubt, by piston speed and the shape of exhaust ports as well as by the brake horsepower of the engine. As these experiments, however, cannot always be made, the author recommends allowing about 0.07 sq. in. of area per b.hp., which figure is the average of the several American and foreign installations. When two headers are used—as on eight or twelve-cylinder engines, it should be kept in mind that each header handles only half of the total exhaust. For most engines this allowance will be found to be liberal.

Exhaust headers have been developed, by experiment, which have had areas as low as 0.045 sq. in. per b.hp., but it would not seem advisable in the absence of a series of experiments to attempt to reduce the figure given.

In cases where headers are inclined to overheat, it has been suggested that they be furnished with a series

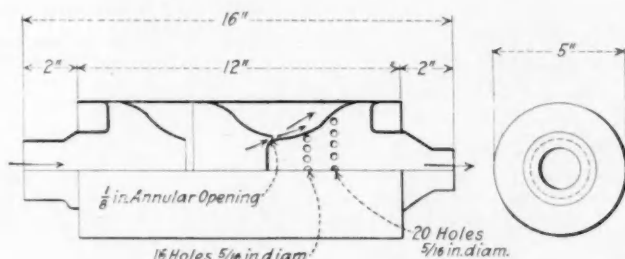


FIG. 14—MUFFLER TESTED AT CORNELL UNIVERSITY

of longitudinal fins, welded to the tube, to increase the radiation. Automobile engines have been built which

were equipped with headers having fins somewhat similar to these.

Table II gives the required outside diameter of No. 20 U. S. gage tube for various horsepower, based upon the figure recommended above. This table, while giving satisfactory results, should not be considered final. It

TABLE I—TESTS ON MUFFLER ILLUSTRATED IN FIG. 14

BRAKE HORSEPOWER		Horsepower Loss, per Cent	BRAKE MEAN EFFECTIVE PRESSURE, LB. PER SQ. IN.		Brake Mean Effective Pressure, Loss, Lb. per Sq. In.	Speed Drop, per Cent
Without Muffler	With Muffler		Without Muffler	With Muffler		
69.8	68.6	1.5	87.7	86.8	0.9	0.52

would be advisable to remove the manifold and note if there is any change in maximum speed of engine when running without it before the question is considered settled for the particular design.

NOTES ON MUFFLER DESIGN

A very interesting and instructive treatise on the principles underlying the design of mufflers is found in part of the previously mentioned report of Profs. H. Diederichs and G. B. Upton.

Another interesting report of experiments, previously referred to, published in *Horseless Age* (d) contains conclusions from results and discussion of the principles of design.

The chief principle to be kept in mind in designing exhaust mufflers is the slowing down of the exhaust gases until their speed is below that of sound. This is accomplished by regular expansion, surface friction and the changing of the direction of flow. Most mufflers embody all three principles in greater or lesser degree.

It would appear to be a reasonably safe assumption that mufflers of the type shown in Fig. 14 may be designed for engines of any horsepower by making the areas of pipes and volumes of chambers proportional to

TABLE II—DIAMETERS OF EXHAUST PIPES FOR 30 TO 300 B. HP. Based on area of 0.07 sq. in. per b.hp.

Brake Horsepower	Area Required, Sq. In.	DIAMETER TO NEAREST 1/16 IN. WITH NO. 20 U. S. GAGE WALL		Brake Horsepower	Area Required, Sq. In.	DIAMETER TO NEAREST 1/16 IN. WITH NO. 20 U. S. GAGE WALL	
		Inside	Outside			Inside	Outside
30	2.1	1 1/8	1 11/16	170	11.9	3 3/8	3 11/16
40	2.8	1 3/8	1 13/16	180	12.6	4	4 1/16
50	3.5	2 1/8	2 11/16	190	13.3	4 1/8	4 3/16
60	4.2	2 3/8	2 3/4	200	14.0	4 3/8	4 3/16
70	4.9	2 1/2	2 5/8	210	14.7	4 7/8	4 3/16
80	5.6	2 5/8	2 3/4	220	15.4	4 7/8	4 3/16
90	6.3	2 3/4	2 3/4	230	16.1	4 7/8	4 3/16
100	7.0	3	3 1/16	240	16.8	4 7/8	4 3/16
110	7.7	3 1/8	3 1/16	250	17.5	4 7/8	4 3/16
120	8.4	3 1/8	3 1/16	260	18.2	4 7/8	4 3/16
130	9.1	3 1/8	3 1/16	270	18.9	4 7/8	4 3/16
140	9.8	3 1/8	3 1/16	280	19.6	5	5 1/16
150	10.5	3 1/8	3 1/16	290	20.3	5 1/8	5 1/16
160	11.2	3 1/8	3 1/16	300	21.0	5 1/8	5 1/16

NOTE.—While the allowances given are liberal and based on experience, this table should not be considered final. Additional experiments to determine effects of changing area are always desirable.

the horsepower. The muffler shown is drawn to scale and was tested on an engine of 60 to 70 brake horsepower.

The Curtiss type muffler seems to be one of the simplest, and most effective devices in use, and some experi-

ences of the author and his associates with mufflers of this type may be of interest. Experimental mufflers were designed along these lines and were tested on engines of different makes. The noise of the exhaust was, in each case, reduced to very little more than the noise of an automobile truck engine of large power. Tests made with and without the muffler in place showed that no difference in the number of revolutions per minute of engine could be detected with the tachometer while the engine was running at full speed. With the engine running at full speed one's hand could be held against the muffler slot without discomfort, in contrast with which it was noticeable that the heat of the gases became uncomfortably great when one's hand was held a couple of feet away from the end of the exhaust pipe without the muffler. The speed of the gases was reduced to a slight puff.

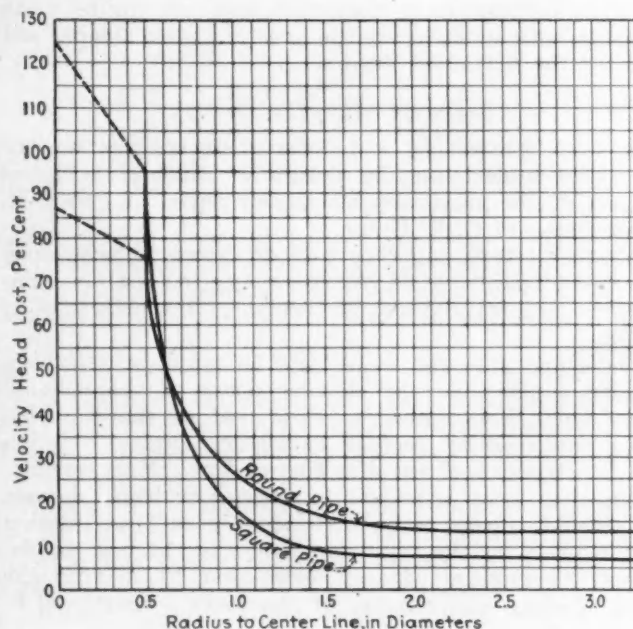


FIG. 15—PRESSURE LOSSES IN PIPE ELBOWS

Other work prevented the carrying on of further experiments, but the results appeared to suggest that the mufflers which were from four to five feet long could have been reduced in length without loss in speed of engine or muffling effects.

While these mufflers were in use, several things were noted which called for attention. One engine was of the geared-down type, and when the muffler was in place the sound of the gears appeared to be of considerable volume, this noise being plainly heard above the sound of the exhaust. As these gears were lubricated by the engine oiling system, it would appear that some reduction

of this noise could be obtained were the engine designed to permit lubricating them with a light grease instead of oil. It was also noted, with most of the engines, that once the exhaust was muffled, several other noises became apparent. These included the rattle of valve mechanism and noise of the propeller. In flight, while gliding with the engine running slowly, the "singing" of the wires, was naturally more noticeable than before. The engines used in the above tests included a Sturtevant 140 b.hp., Hall-Scott Model A-5-A, Liberty 8 and Liberty 12.

No high degree of accuracy is claimed for the tests described, as they were field and not laboratory tests. The results indicated, however, that in design of mufflers of this type it is advisable that the slot be kept narrow (1/16 in. was found to be satisfactory), and that a net area of 0.05 to 0.06 sq. in. per b.hp. is ample for it. It did not appear to be necessary to leave any opening at the rear end as was done in the Curtiss R-4 muffler, better results in muffling having been obtained with the smallest end. It is interesting to note that the mufflers now used on the Curtiss JN Hispano Suiza model have only a small opening in the end.

MATERIALS OF CONSTRUCTION

The materials of construction naturally depend upon the type of machine and its size, but for all ordinary work the use of No. 20 U. S. gage for headers and No. 22 U. S. gage for pipes and mufflers will insure parts which will hold up well. For machines of which extreme climbing ability and consequent low weight are demanded, these could be reduced to paper thickness, but it would probably be necessary to renew them often.

Although contrary to general opinion, the author has been informed, on good authority, of cases in which the horsepower of an engine with open exhausts has been increased by fitting it with a well-designed manifold. This was found in some cases by dynamometer tests and in others by noting the r.p.m. for the two conditions. This effect is, no doubt, due to the gases moving at such high speed that suction is set up in the header, thus giving the effect of exhausting into partial vacuum.

REFERENCES

- (a) Report No. 10 in Reports of U. S. Advisory Committee for Aeronautics, 1916-17.
- (b) Data sheets issued by the American Blower Company.
- (c) U. S. Patent No. 1,106,193 to Deperdussin.
- (d) Automobile Muffler Losses Experimentally Determined, *Horseless Age*, May, 1916.
- (e) Heat-balance Tests of Automobile Engines, by Prof. W. T. Fishleigh and W. E. Lay—*Transactions S. A. E.*, 1917, Part I, page 82.
- (f) *Fan Engineering* (1914 Edition) by Willis H. Carrier.
- (g) Loss of Pressure Due to Elbows in Transmission of Air Through Pipes and Ducts, by F. L. Busey, *Trans. Amer. Soc. of Heating and Ventilating Engineers*, 1913.
- (h) Flow of Air in Heating and Ventilating Ducts, by L. A. Harding—*Trans. American Society Heating and Ventilating Engineers*, 1913.
- (i) Coefficient of Friction of Air Flowing in Round Galvanized Air Ducts, by Prof. J. E. Emswiler—*Trans. American Society Heating and Ventilating Engineers*, 1916.



I—Modern Aeronautic Engines

By HERBERT CHASE* (*Member of the Society*)

SEMI-ANNUAL MEETING Report

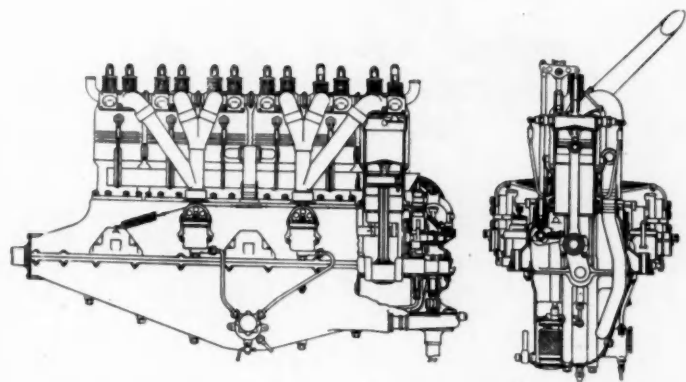
Illustrated with CHARTS AND PHOTOGRAPHS

OBTAINING data concerning war equipment is no easy task in war time and is beset with special difficulty when the purpose of compiling the data is to make it available in a public meeting. Most of the data to be presented here were obtained by others to whom credit would be given if names could be mentioned.

Under the circumstances I can attempt only to point out a few features of engines, some of which are obso-

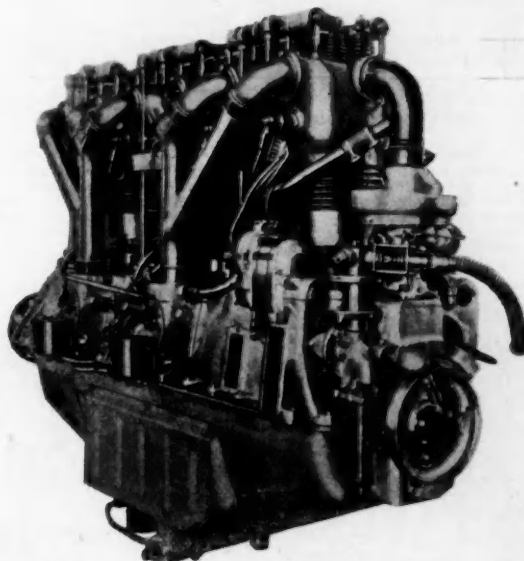
lete, and a few of which are not yet beyond the development state. Some, however, are representative types and are worthy of careful study. No attempt to describe the rotary cylinder type, or the radial cylinder type, has been made, nor are any air-cooled types discussed. The description is in two parts; one is given here and the other will appear in a later issue.

*Assistant Secretary, Society of Automotive Engineers, Inc.



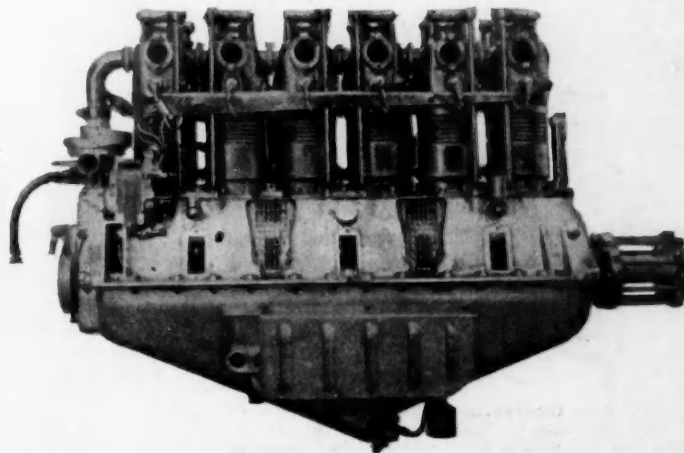
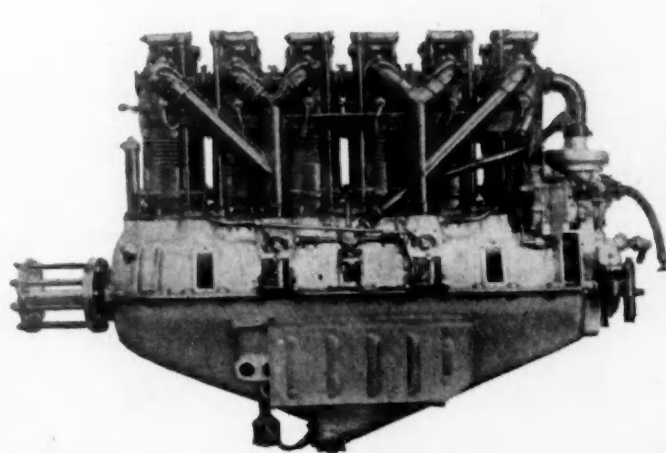
BENZ ENGINE, 160 HP. AT 1400 R.P.M.

Bore 5.12 in., stroke 7.09 in.—six cylinder. Compression ratio 4.5. Weight dry 591.5 lb., or 3.7 lb. per b.h.p. Features same as in larger size except that two valves per cylinder instead of four are used. This was one of the most successful early aviation engines and won Kaiser's prize, prior to the war.

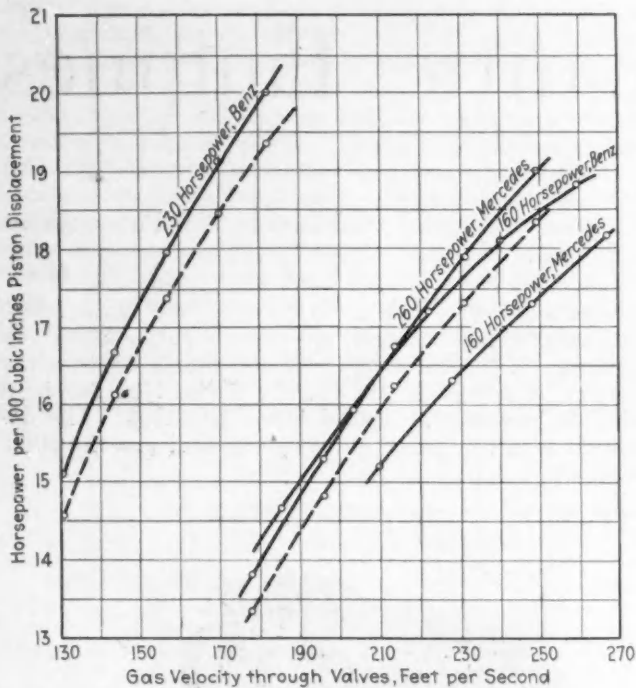


BENZ ENGINE, 230 HP. AT 1400 R.P.M.

Bore 5.71 in., stroke 7.48 in.—six cylinder. Compression ratio, 4.91. Weight dry 848.3 lb., or 3.68 lb. per b.h.p. Note position of pump which is mounted above crankcase. Most of water is discharged across top of the cylinder close to exhaust valves, but some is discharged at base of cylinders. Louvers catch air which is forced through tubes in crankcase to cool oil, and out through louvers on other side. Carburetor is fastened against crankcase, air being taken through crankcase.

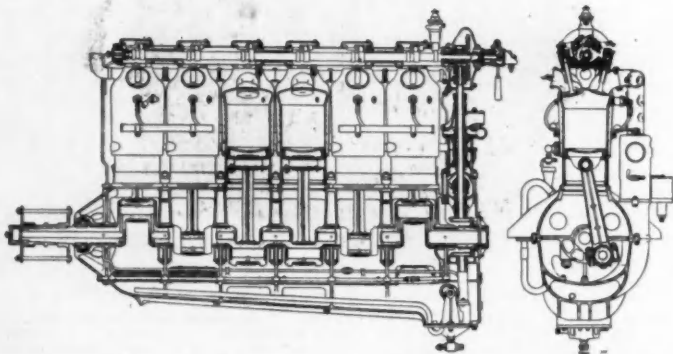


BENZ 230 HP. SHOWING AIR INLET FOR CARBURETOR AND OUTLET FOR OIL COOLING AIR. NOTE WATER CONNECTIONS BETWEEN BASES OF CYLINDERS



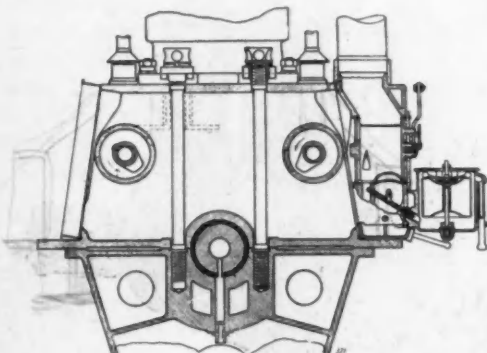
GAS VELOCITIES IN BENZ AND MERCEDES ENGINES

Curves showing how the 230-hp. Benz, with manifold of such relatively small area as to give gases same velocity as they have through valves, gives a greater power per unit of displacement than the smaller Benz, or the Mercedes, which has a much larger manifold and correspondingly low gas velocities therein. In the large Benz, however, a radical timing is used, the inertia of gases in manifold apparently being depended upon to fill cylinders.



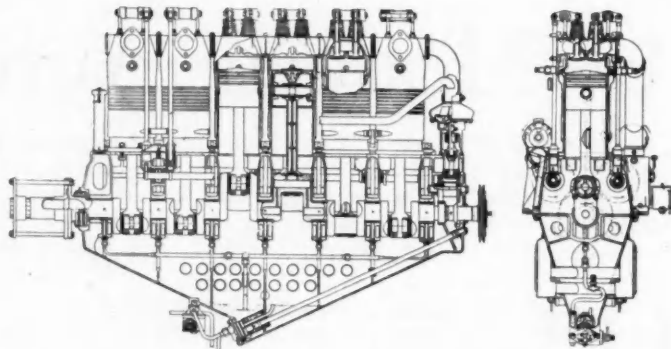
MERCEDES 160 HP. AT 1400 R.P.M.

Bore, 5.51 in.; stroke, 6.3 in.—six cylinder. Compression ratio, 4.5. Weight dry, 618 lb. or 3.8 lb. per b.h.p. Two valves per cylinder. Light pressed-steel inlet and exhaust elbows welded into steel cylinders; these apparently have given trouble by burning out, since a heavier drop-forged type is now used. This is the prototype of most modern steel cylinder constructions. Exhaust valve stem is entirely surrounded by jacket water. Built-up piston with drop-forged head. Cast iron skirt welded on. The pin bosses are carried in head portion.



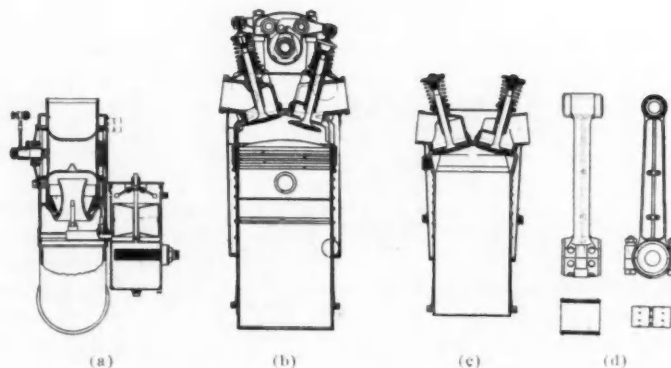
BENZ CARBURETOR, WATER-JACKETED

Separate leads to pump to insure circulation. Starting and running floats controlled by barrel-type throttle. Through bolts into lower half of crank-case.



BENZ 230 HP., SHOWING SECTION THROUGH OIL TUBES

Pressure feed oil on all bearings including piston-pin. Note use of ball-bearing rocker shaft, which makes it unnecessary to enclose, and to lubricate these bearings by force feed. Bushing is used on upper part of valve stem only—apparently to permit lower part to be more effectively cooled. Tappets are of very generous size. Four valves are used, these having the following radical timing: Exhaust opens 60 deg. early; Inlet closes 55 deg. late.



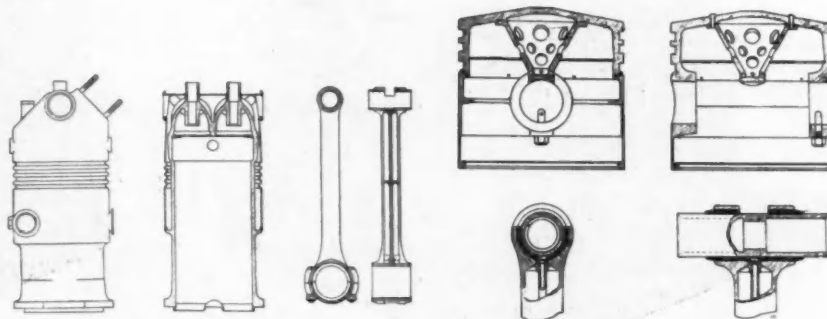
PARTS OF MERCEDES AVIATION ENGINE

(a) Carburetor—probably largest single carburetor used. Has ball-bearing throttle, water jacketed; this construction probably is used to prevent freezing. The starting jet is located above throttle. A movable throat bushing, which rises as the air speed through carburetor increases, is used.

(b) Latest type Mercedes steel cylinder. Has drop-forged head screwed and welded on to a tubular sleeve. Drop-forged elbows are welded in. Only one side of the exhaust-valve stem is adjacent to jacket, a departure from earlier practice in which stem guide was completely surrounded by water; water is supplied direct from pump to vicinity of exhaust valve as well as base of cylinder.

(c) Steel cylinder in intermediate state of development. One piece with welded-in drop-forged elbows.

(d) Section of rod showing oil pipe to piston pin.

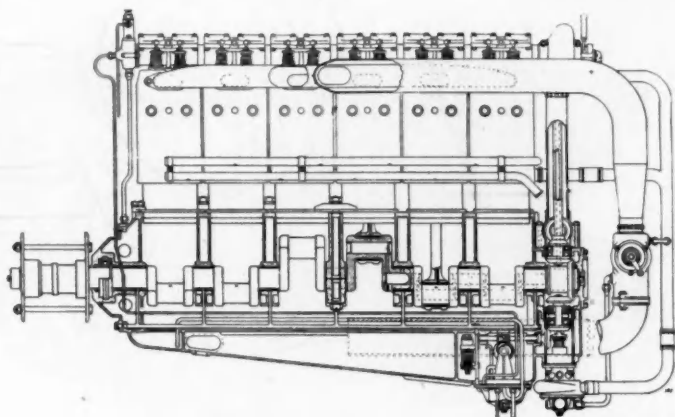


BENZ CYLINDERS, CONNECTING-RODS AND PISTONS

Cast iron cylinder of 160 hp. (two valve) Benz. Note welded steel jacket, and four bolt connecting-rod. Note unusual construction of pistons. Material is cast steel (drop forged) pillar, riveted and welded to head. This carries load from head direct to bearing and also dissipates some heat to piston-pin bearing. No scraper ring is used but oil drip is provided. The piston-pin is not of uniform section.

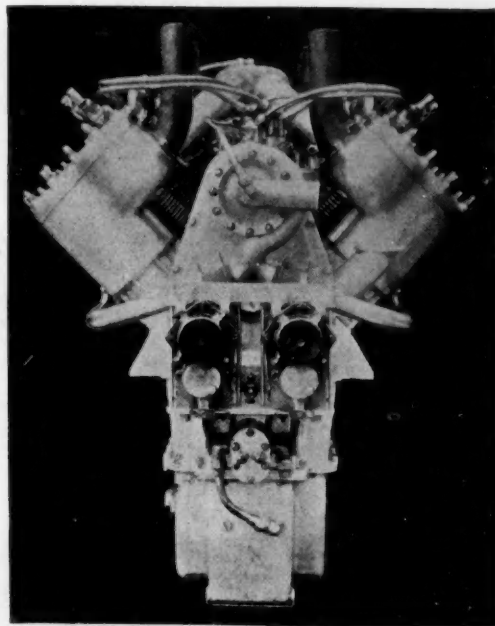
MODERN AERONAUTIC ENGINES

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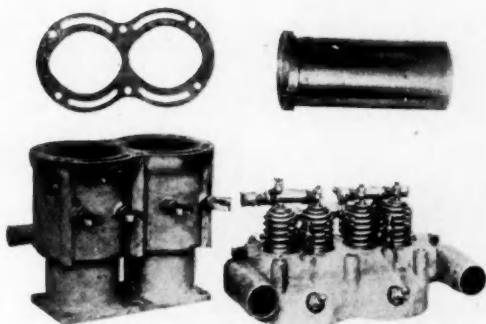


MERCEDES 260 HP. ENGINE

Part of inlet manifold is concentric with inlet pipe, the latter being of very large diameter. The camshaft drive runs at twice crankshaft speed. Force feed lubrication is provided for all bearings including piston pin. Oil is also led to the center of the overhead camshaft and returns through hollow drive shaft to crankcase. Lower main bearing caps are fastened by through bolts to yokes holding down the cylinders.

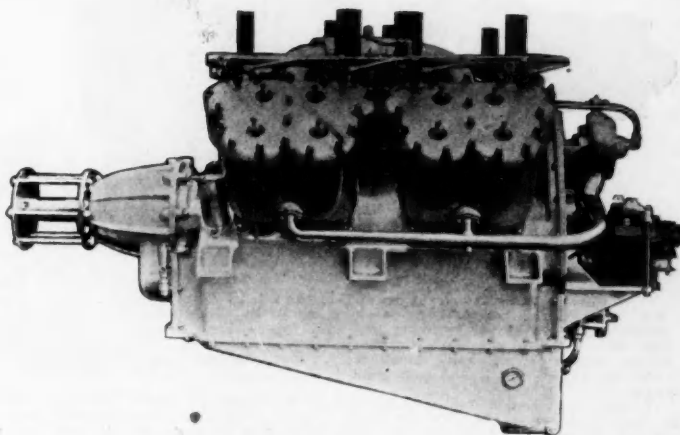


THOMAS ENGINE (END VIEW), SHOWING EXHAUST PIPES IN CENTER, CLEAR OUTSIDE



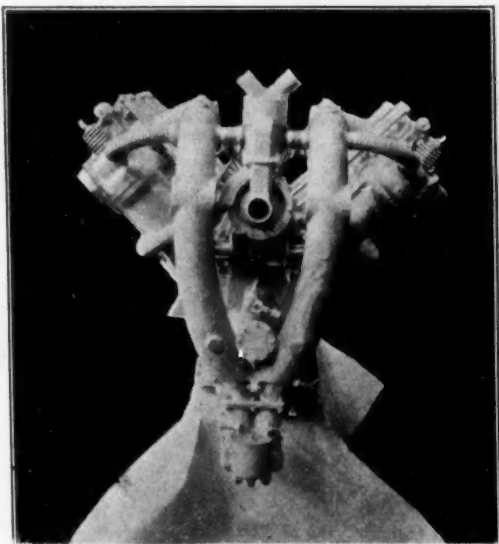
STURTEVANT CYLINDER BLOCK

Showing sleeve with flange at top, and separately cast head; a good construction for rapid production and ready repair. Does not require removal of cylinder to grind valves.



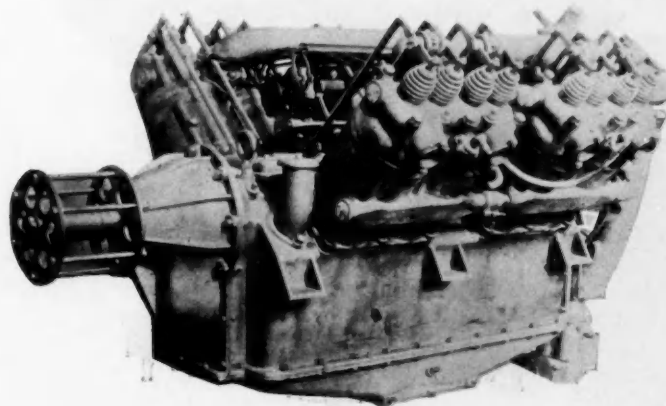
THOMAS ENGINE, 135 HP. AT 2000 R.P.M.

Bore, 4 in.; stroke $5\frac{1}{2}$ in.—eight cylinder, 90 deg. V. Weight dry with starter, 650 lb., or 4.66 lb. per b.h.p. High speed geared down. L-head—separately cast heads, two cylinders per block.



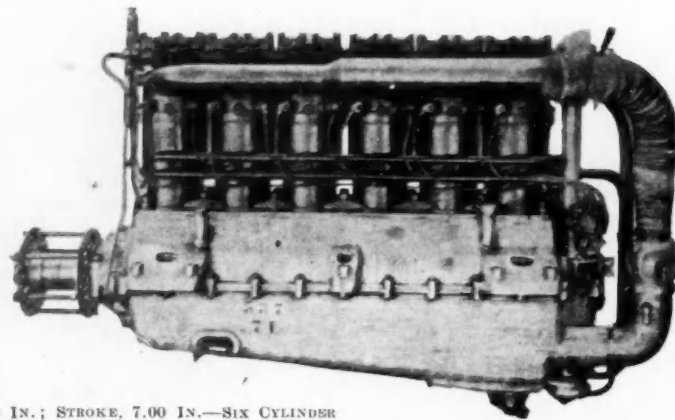
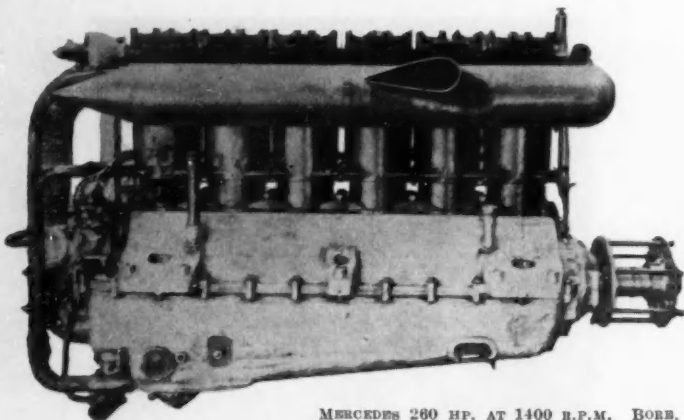
END VIEW OF STURTEVANT ENGINE

Showing long jacketed manifold gravity feed to carburetor. Flexible metal hose conveys water to distributing head, where a thermostat is employed to regulate temperature.



STURTEVANT ENGINE, 200 HP. AT 2250 R.P.M.

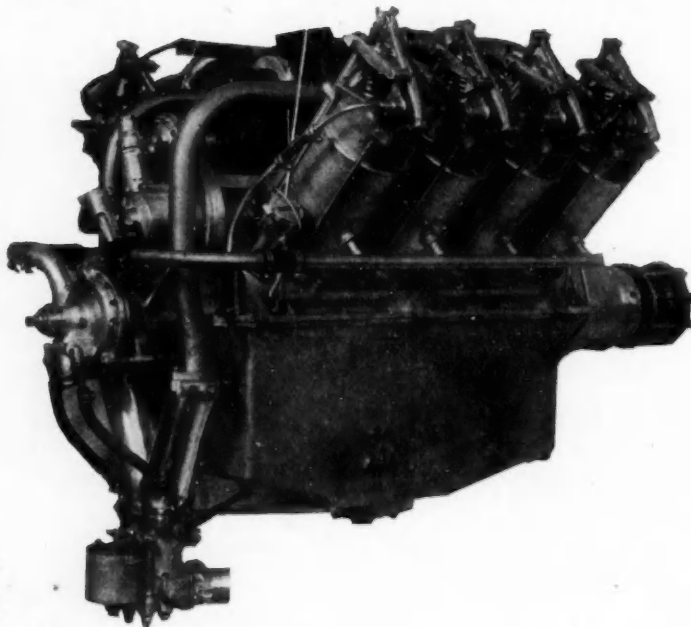
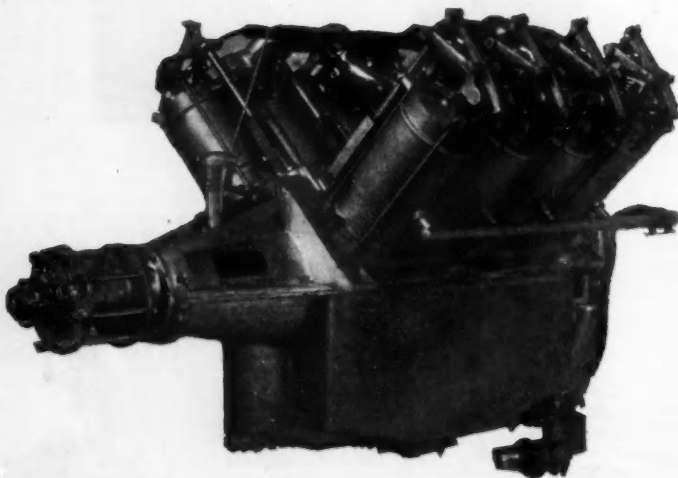
Propeller speed, 1350 r.p.m.; bore, $4\frac{1}{4}$ in.; stroke, $5\frac{1}{2}$ in. Weight or 2.3 lb. per b.h.p. Compression too high for wide open operation level. Overhead valves operated by push rods from crankcase. From cylinder head to crankcase are employed.



MERCEDES 260 HP. AT 1400 R.P.M. BORE, 6.00 IN.; STROKE, 7.00 IN.—SIX CYLINDER

Compression ratio, 4.94. Weight, 936 lb., or 3.71 lb. per hp. Showing water-pump located below crankcase, with water circulated direct to head adjacent to exhaust valve and to base of cylinder with branch to carburetor

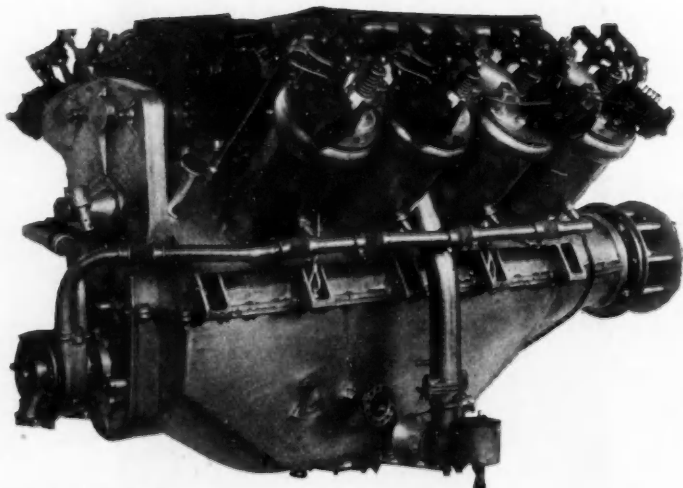
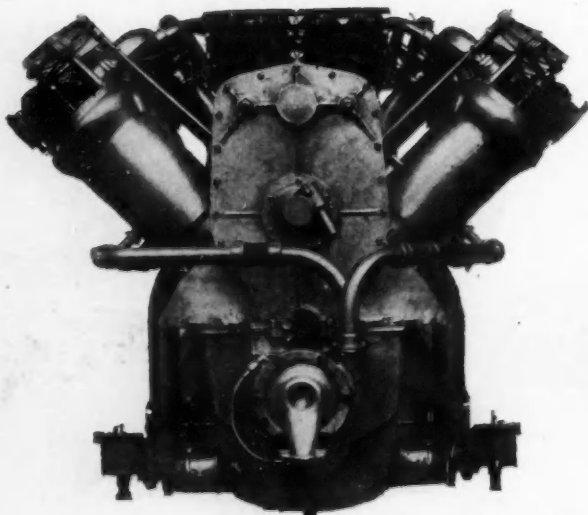
jacket. The carburetor is placed rather low and the air drawn into it enters the lower pan and circulates under the oil pan. One spark-plug is placed under each inlet valve. There are two inlets and two exhaust valves per cylinder.



CURTISS OX-5 90-HP. EIGHT-CYLINDER ENGINE

Normal r.p.m., 1400. Bore and stroke, 4 by 5 in. Arrangement, 90-deg. V. Compression ratio, 5. Cooling, water by centrifugal pump on end of crankshaft. Valves per cylinder, one inlet; one exhaust in head. Weight dry, 384 lb.—4.26 lb. per hp. Cylinders, single, cast iron, with Monel metal jackets brazed on. Pistons, aluminum. Connecting-rods, side by side, plain. Valve mechanism, push and pull rods to rocker arms. Ignition, high tension magneto.

Carburetion, Zenith duplex carburetor at the front end feeding through long pipes to a two branch pipe, each branch of which feeds two cylinders. Oiling, pressure feed into camshaft from sump, thence metered through camshaft bearing to the main bearings, thence to connecting-rod bearings through the hollow crankshaft, and then back into the sump through a space in the oil partition. Splash feed to piston and pins.



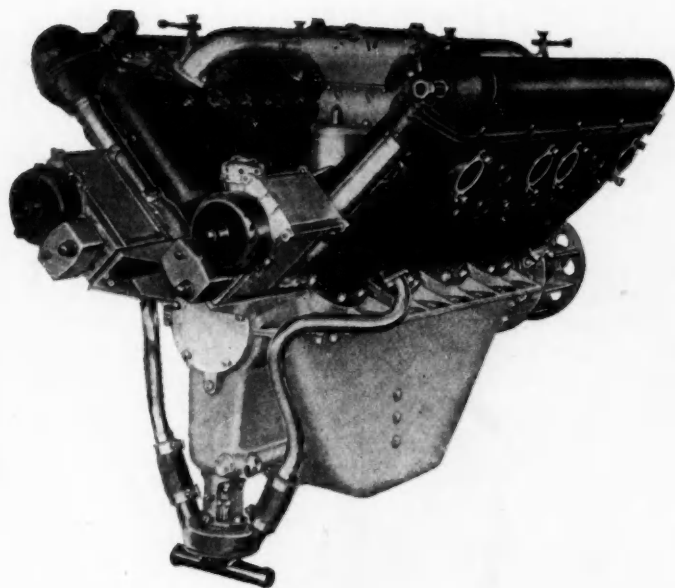
CURTISS V2 200-HP. EIGHT-CYLINDER ENGINE. NORMAL R.P.M., 1400. BORE AND STROKE, 5 BY 7 IN.

Arrangement, 90-deg. V. Compression ratio, 5.0. Valves, two in head; air-cooled stems. Seat in water-cooled steel cylinder. Cylinders, single, steel, with Monel metal jackets brazed on. Weight dry, 700 lb. Pistons, aluminum. Connecting-rods, plain, side by side. Valve mechanism, push rods and rocker arms from single camshaft. Ignition, two high tension magnetos. Carburetion, two Zenith, single carburetors on long water-jacketed induction pipes. Oiling system, pressure feed to inside of camshaft, from the oil sump, thence

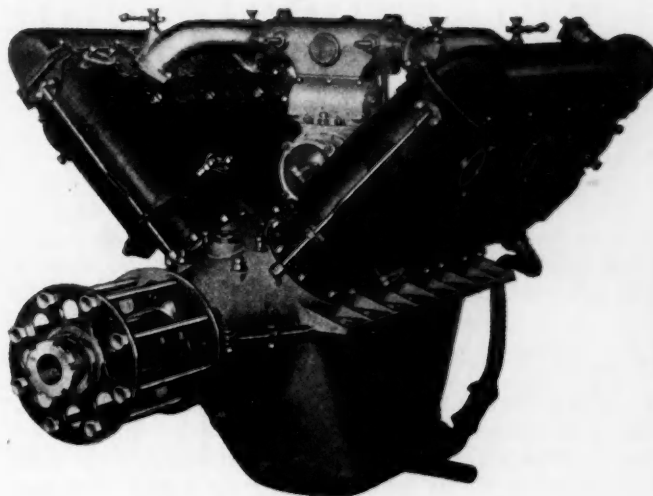
metered through the camshaft bearings to the main bearings, thence through the hollow crankshaft to the connecting-rod bearings, thence through an oil tube to the piston pins, thence to the cylinder walls, thence back to the oil-pan partition from which it is sucked by two suction pumps back into the oil sump. Cooling system, water cooled by centrifugal water-pump geared to crankshaft. Valve port air-cooled, exhaust having cooling fins.

MODERN AERONAUTIC ENGINES

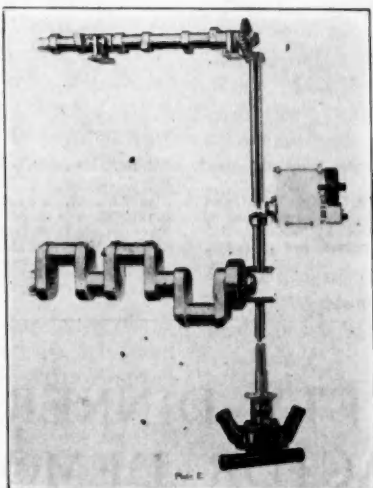
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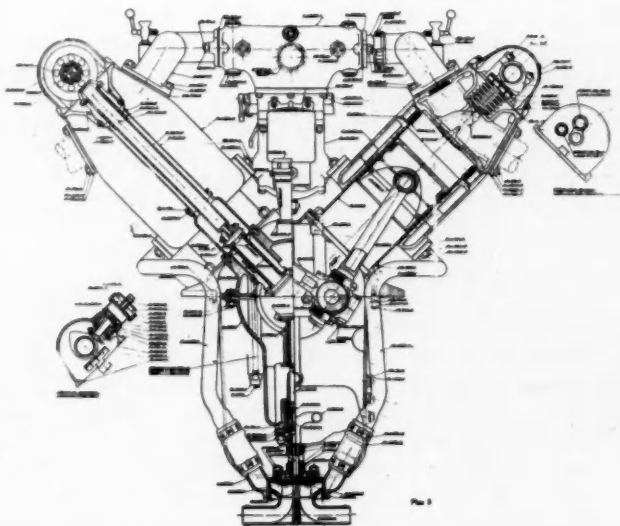
HISPANO-SUIZA. 150 HP. AT 1450 R.P.M. BORE, 4.72 IN.; STROKE, 5.11 IN. EIGHT CYLINDER, 90 DEG. V. COMPRESSION RATIO, 4.7. WEIGHT DRY, 442 LB., OR 2.95 LB. PER H.P. CLOSED STEEL SLEEVE WITH FLAT HEAD IS SCREWED INTO ALUMINUM BLOCK CASTING, THE VALVES SEAT IN SLEEVE HEAD. THE LATTER IS FLANGED AT BASE, WHERE IT IS BOLTED TO CRANKCASE



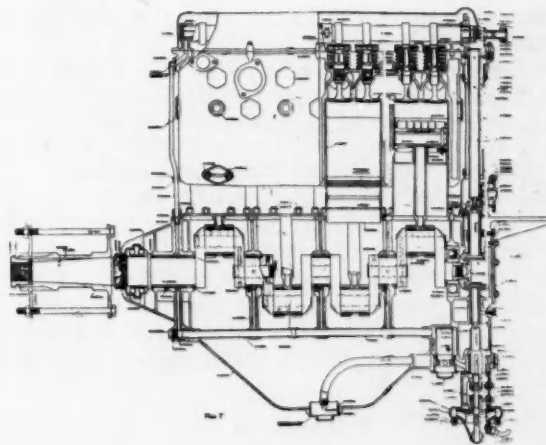
HISPANO-SUIZA. ALUMINUM BLOCK CASTING IS ENAMELED INSIDE AND OUTSIDE; THE ENAMEL BEING BAKED ON. CARBURETER IS LOCATED IN V, GIVING SHORT MANIFOLD. THE LATTER IS WATER JACKETED. ENGINE IS CLEAN CUT DESIGN, ENABLING GOOD STREAM-LINING. WATER CONNECTIONS ARE COPPER PIPE.



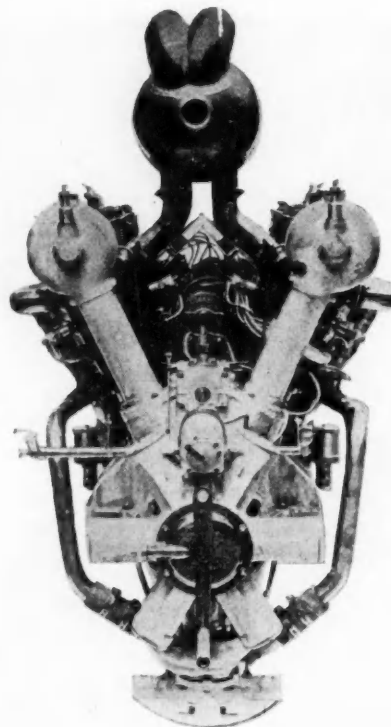
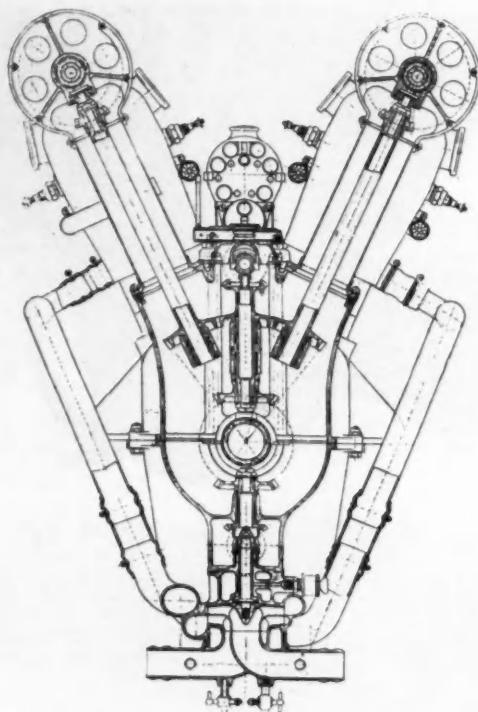
SIMPLE AND COMPACT GEAR TRAIN USED ON
HISPANO-SUIZA ENGINE



HISPANO-SUIZA. VALVES ARE OPERATED BY CAMS DIRECTLY OVER AND BEARING ON VALVE DISKS, NO ROCKER ARMS BEING REQUIRED. ALUMINUM PISTONS ARE RATHER SHORT, AS ARE ALSO THE CONNECTING RODS



HISPANO-SUIZA. CRANKSHAFT HAS FIVE MAIN BEARINGS. FORCED FEED LUBRICATION IS PROVIDED TO ALL BEARINGS, INCLUDING THOSE OF CAMSHAFT, PISTON-PINS AND PISTONS ARE SPLASH LUBRICATED. THE OIL AND WATER PUMPS ARE LOCATED ON THE SAME SHAFT



END VIEWS OF RENAULT 400 H.P. AVIATION ENGINE. BORE, 4.92 IN.; STROKE, 6.3 IN. TWELVE CYLINDER, $47\frac{1}{2}$ DEG. V.

Weight, 800 lb., or 2 lb. per b.hp. Steel cylinders, with inlet and exhaust elbows welded in. A welded-on jacket for each pair of cylinders is employed. The exhaust valve guide is completely surrounded by water. Exhaust valves are inside, adjacent to the V. Two 2 $\frac{1}{2}$ -in. valves per cylinder,

also two spark-plugs per cylinder, so placed to be easily accessible. Four magnetos are used for ignition. Magnetos and exhaust manifold in center. The camshaft drives run at three times camshaft speed. The hand magneto used in starting and air starter, driven by over-running clutch, are shown.

SOCIETY DINNER AT SALINA TRACTOR DEMONSTRATION

AT THE time this issue of THE JOURNAL goes to press, arrangements have practically been completed for a Society dinner to be held at the Hotel Lamer in Salina, Kan., during the National Tractor Demonstrations. The dinner is scheduled for the evening of July 31, although the demonstrations will last during the week from July 29 to Aug. 3 inclusive.

The demonstrations are being held under the auspices of the Tractor and Thresher Department of the National Implement and Vehicle Association.

Since it is the only tractor demonstration of the year lasting an entire week, it is expected that practically all of the manufacturers will be represented and will display tractors and all other kinds of farm apparatus. The usual plowing work will be done and arrangements are being made for other tests which may be entered into voluntarily by exhibitors.

Further details regarding the dinner and the addresses on tractor subjects that will be delivered there will be given in the next issue of THE JOURNAL.



Development of Gun Manufacture

By LIEUT. W. H. W. SKERRETT* (Non-Member)

DETROIT SECTION PAPER

Illustrated with CHARTS

BEFORE the beginning of the Great War, the average civilian had but a vague idea of the part which artillery plays in the modern battle. The infantryman with rifle or revolver was the most familiar figure in army life; he and the sabered cavalryman typified the fighting man and the spirit of war. It was not until September, 1914, after Liège and Namur had fallen in rapid succession and giant 42-cm howitzers of the Germans were being turned against the thin line of British troops in France, that the importance of artillery dawned upon the public.

As we look back upon those terrible days which preceded the battle of the Marne and the Aisne, we realize how grateful civilization must be that the English and French had foreseen the rôle which field artillery would play in a modern battle. On those September days when the fate of civilization hung trembling in the balance, the waves of massed Germans were broken and flung back from Paris by thin lines of men and of guns. Only a handful of British troops, compared with the armies now in the field, by sheer grit held the Hun at bay until the French could gather their men for the blow which stopped the German advance and made the enemy dig in on the Marne, and later drove him out of his trenches and made him retreat. Most of us then began to ask for information about guns and ammunition.

We have heard of drum fire, barrages and a dozen other features of artillery work, we have read of gun fire from the Flanders battle front being audible across the English Channel at Dover and other coast towns, of artillery preparation so complete that the craters formed by the bursting shells overlapped. An instrument which can project a mass of explosives and metal weighing a ton at the rate of 2200 ft. per sec. with such accuracy that six shells form overlapping craters at a range of 17 miles is an engineering creation which demands our attention. Another type is able to discharge 30 shells per min., each one aimed and timed to do the greatest damage to hostile troops.

A series of these can project a wall of leaden bullets and flying fragments of steel of such density that nothing can live through it if exposed to its direct force—a spray of steel which covers the ground beneath it with metal sufficient to disable anyone who tries to advance through it, yet permitting those who are depending on it for defense to walk behind it with a sense of security knowing that its accuracy is 99.9 per cent perfect, even though the gunners directing the fire are unable to see the object upon which they fire.

When one realizes that some 600 guns of all calibres have been concentrated upon a strip of territory less than three miles long and about two miles wide, some idea of the intensity of fire resulting may be obtained. On a front of 3½ miles at Verdun the Germans had approximately 150 batteries of all sizes while on a front of 2½ miles the French had about 120 batteries of guns of all sizes. During the height of the storm for the pos-

session of some of the key fortresses in the line as many as 1,000,000 shells per day were fired, the number being about equally divided between the French and the Germans.

Early Types of Cannon

The original gun bears as little resemblance to the pieces used today as the original "horseless carriage" with its sleigh front and handle-bar steering arrangement resembles the modern sedan with all its improvements and refinements of body and powerplant.

The first guns of which we can find record appear to have been made in Ghent in 1314, at which time an entry was made in the town clerk's records of a "gun with powder" being sent to England. According to pictures made about that time, the gun was vase-shaped (the Italian name for it was "vasi") and was used for shooting darts. Powder was placed in the bowl of the vase and an iron dart was rammed firmly into the neck. The neck was made gas-tight by means of a tampon wound around the shaft. A touch hole in the side was used. The range was a few hundred yards and the damage done by the dart was considerable, although the noise of the explosion seems to have been more effective than the missile, for a historian of that time tells us that "the thunder struck terror to men and horses so that they dared not fight"—and, as an afterthought, "some there were who suffered wounds."

Most of us believe that guns were first used in 1346 at the Battle of Crecy, for it was there that the English used their three guns with great success against foot soldiers and mounted men. This is perhaps the first record of guns in a field battle, and so we may say that field artillery was born at Crecy, although fifteen years before (1331) the Germans had used siege guns at Clivdale in Italy. The projectiles used were rounded stones smaller in diameter than the bore of the guns. Pressure was obtained by wadding the charge with turf. The guns at Crecy used iron slugs, which were very effective against horses.

Little development occurred in field artillery, however, during the next hundred years, although siege guns of large size appear to have been used. In 1446 the hand gun or musket made its appearance and became the deciding factor in battles from that date. Owing to their lack of mobility, guns were not adapted for field work and were used only during sieges, when the time necessary for transporting artillery from one point to another permitted them to be brought into position; the pieces were clumsy and heavy and the wagons then in vogue were also cumbersome. With the increasing superiority of musket fire, artillery quickly fell into disuse, and not until 1630 did it again advance to a position of importance.

First Use of Mobile Artillery

In 1631 Gustavus Adolphus, King of Sweden, made use of mobile artillery in sweeping over most of northern

*Inspector of Ordnance, Ordnance Reserve Corps, U. S. A.

Europe, especially that part now occupied by the German Empire, and, with less than fifty guns, drove his opponents into their fortified towns where he besieged and defeated them. His greatest enemy, Tilly, with a much superior force, was unable to bring up his heavy guns (24 pounders, which correspond to our 4-in. pieces) and was routed time and again by Gustavus' cast-iron 4 pounders, which were able to pour a heavy fire upon his flanks.

Gustavus was the first to classify and set limits upon heavy and light artillery. He classed as light or mobile artillery all guns up to and including the 12 pounder (a light 3 in.), but recommended an 8 pounder or lighter for extensive campaigning. In fact, in 1632, when he finally defeated Tilly, he was able to cross the Danube in face of a much superior force by reason of the mobility of his 4 pounders, of which he then possessed about sixty.

During the seventeenth century the organization of the artillery was very loose. Apparently the only persons in the military service were the gunners or "masters of ordnance" who supervised the firing of the pieces, laid the guns and took the place of the officers of a modern battery. Their assistants were hired men, who usually deserted or abandoned the guns at the first signs of danger. These men were released as soon as the need of their services disappeared, and, very naturally, the training had to be repeated each time a battery went into the field.

With the introduction of the flint-lock musket, the use of artillery became limited to siege operations, where sufficient time for organizing and equipping a siege train was allowed. Therefore, large calibre guns were constructed with bores of 12, 14 and 16 in. which threw stone or iron projectiles weighing as much as 700 to 800 pounds. Some great pieces of 20-in. bore were also constructed, but difficulties in mounting them arose and the engineers of the times were unable to find a solution.

Burning shells were introduced and a few explosive shells were used.

Advent of Breech-Loading Guns

Some early attempts were made to overcome the difficulties of muzzle-loading guns, and breech-loading pieces were unsuccessfully tried. These were failures for two main reasons; the necessary strength could not be obtained and the gases leaked through the joints. So hazardous, in fact, were these guns that they were abandoned, and it was not until 1845, when Major Cavalli, a Sardinian officer, introduced the sliding wedge type of breech block, that muzzle-loaders began to lose first place.

In this gun, which was also one of the first to be rifled, the chamber was sealed by a copper plug and an iron wedge was inserted through the body of the gun, which prevented the copper from being blown out.

Until 1739 all guns had been either cast or forged on a mandrel, and naturally the bore was rough and full of irregularities, which allowed the gases to leak around the projectile and reduced the power and range to an enormous extent. The range, at best, was only a few hundred yards, and the flight of the shell inaccurate. The wadding was very important, as upon it depended the pressure developed and therefore the velocity of the shell.

Powders, also, were an unknown quantity, some being much more powerful than others. We wonder that results could be obtained when so many variables were introduced.

With the advent of bored guns—Maritz, a Geneva gun-

maker, in 1739, having bored the first from a solid bronze casting—the power and accuracy increased immensely. The shell was made to fit the bore closely and much higher velocities were attained. Aided by the explosive shell, artillery became a factor in battle against which musketry could not compete. The use of bombs and grape-shot against fortifications and troops greatly changed military tactics.

In the eighteenth century wheeled mounts were also introduced successfully, and artillery became mobile enough to accompany troops in the field, to maneuver ahead of them and to break up enemy formations before the infantry attack could be delivered.

Artillery First Used for Defense

During the time of Frederick the Great, the use of artillery underwent a great change. During the early part of the eighteenth century the approved strategy consisted of driving the enemy's army back upon some fortified town by a series of cavalry and infantry attacks or battles, and then besieging the town until it surrendered owing to sickness or starvation. If necessary, heavy howitzers were brought up and the fortifications demolished by gunfire until the chances of carrying the place by assault were assured. Frederick the Great, about 1760, introduced the idea of using artillery as a terrible defensive weapon by luring the enemy into battle and then annihilating him by massed batteries using grapeshot. Thus by the end of the Seven Years' War his "wall of four hundred cannon" was known as the iron ring through which no army could break, and at Burkersdorf he sent a great battery of 50 howitzers into action against the fortifications, which were reduced in two days. Thus siege operations were reduced from months to days and artillery became more important as a deciding factor in battle.

Napoleon used artillery to break up both foot and cavalry charges in both frontal and flank attacks and always maneuvered to bring his artillery out in front of his troops so that the guns might open gaps in the enemy line, through which he could pour his troops. It was not until his opponents learned to follow his example that he was checked.

By the time of the Civil War, the position of artillery was changed again, for the long range musket or rifle had been introduced shortly before and the guns needed infantry support. Therefore the batteries were placed in the same line as the troops and the enemy had to advance into a concentrated rifle and artillery fire in which his own guns could take little or no part due to fear of firing upon their fellow soldiers. The advent of the rifled gun and the explosive projectile made possible the use of batteries behind the lines out of sight of enemy observers, but due to the imperfections of the ammunition, the damage done was not in proportion to the increase of range.

Time fuses became more accurate and shrapnel began to be introduced—the rifled gun increased in accuracy and the breech mechanisms improved.

Gun Stresses

With the demand for greater range and accuracy, attention turned to increasing the strength of the pieces. It was recognized that a gun was subjected to two strains—one circumferentially, which tended to split the barrel longitudinally, and a second which acted to break the gun apart lengthwise by blowing off the breech or muzzle end. The circumferential stress is the most difficult to resist, for the strength of a simple gun, that is, a solid one-

piece gun, soon reaches a limit of thickness beyond which additional metal gives practically no increase in strength to resist circumferential stress. In a simple gun, only extension of the metal occurs, and when it has reached its limit, rupture occurs. If a gun is built up upon a system of initial tensions a compression will result in the metal of the bore, which compression must be overcome before the extension of the metal can occur. Thus the strength of a simple gun is proportional to the ex-

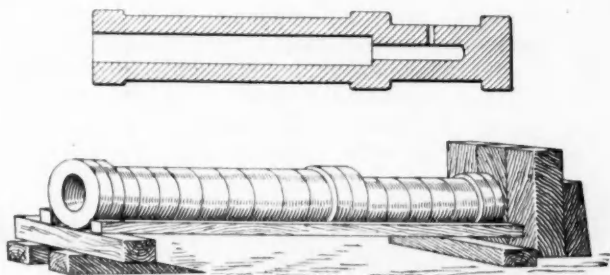


FIG. 1—TYPICAL GUN OF THE FOURTEENTH CENTURY

tension that it can withstand before rupture, while a built-up gun is proportional to the compression plus the extension.

Rifling of Guns

Lord Armstrong, in 1855, introduced the first built-up gun of this type and met with great success. Wire winding was introduced at the same time and was also successful from the start. From 1850 to 1870 the main improvements were in breech mechanisms, rifling and ammunition. The importance of steady flight was recognized, and the long tapered shell took the place of the round shot previously used. This shell was given rotation in various ways, the first plan being that of a twisted ellipse or hexagon. The Whitworth gun had a cross-section which was hexagonal and a twist of one turn in 25 calibres, and the shell was cast with a similar form, making a good mechanical fit in the bore. This gun and the Lancaster oval bore gave trouble by the projectile becoming wedged. Upon this occurring the gun would explode, so that they were abandoned in favor of the grooved rifling which was adapted to studs or lead and copper bands. The copper driving band is used at present. The band is slightly larger than the diameter of the gun, so that the soft copper is forced into the grooves, forming a gas-tight joint, and the twist causes the projectile to rotate, tending to steady its flight. The rifling is usually made with an increasing twist. In 1880 the metallic cartridge case was introduced.

Non-Recoiling Carriages

In 1890 the non-recoiling carriages were used. Previous to this the gun and carriage as a whole recoiled after each shot and had to be realigned before firing again. Also, any attempt to limit its recoil had been made by increasing the weight of the mount. The constraining of recoil developed great stresses in the carriage and made heavy cumbersome mounts necessary. With the spring recoil and non-recoiling carriage, the gun was automatically brought into the firing position after each shot. This allowed the carriage to be aligned, and the alignment was held until firing in that position was discontinued. It made rapid fire possible and simplified

breech mechanisms, and fixed ammunition in metallic cases further increased the rapidity with which the guns could be discharged.

Accuracy of Shrapnel Fire

Time fuses were perfected and shrapnel became one of the most used types of projectile. It is now possible to lay a barrage of shrapnel fire that is accurate to 50 yards, a thing which was impossible until just before the beginning of the present century.

The power and velocities of guns were also increased until today we have a gun with a velocity of 4000 ft. per sec. which shoots more than 70 miles.

Development of Early Guns

We will now consider the component parts of a simple gun and then trace the successive stages of development through which it has passed in the last few decades, for since 1850 the strides have been enormous.

Fig. 1 shows a simple gun such as was used in the fourteenth century. It consisted of a cast-iron barrel slightly larger than the diameter of the shot which it expelled and a small powder chamber which was filled from the muzzle. The gun was held rigidly between two timbers and prevented from recoiling by stakes driven into the ground. It was elevated by means of wedges placed beneath the muzzle and had a range of a few hundred yards.

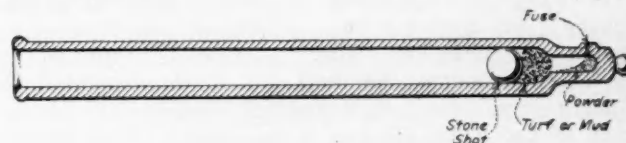


FIG. 2—SECTION OF EARLY GUN

Fig. 2 shows a later development of the same type, which was the kind used for siege purposes during the fifteenth century. These guns often had a bore as great as 20 in. and fired a stone missile weighing 700 lb. a distance of 400 yards. The gun itself weighed 14,000 to 16,000 lb. without its mount and was not mobile. Often these guns were set permanently in emplacements and kept loaded at all times.

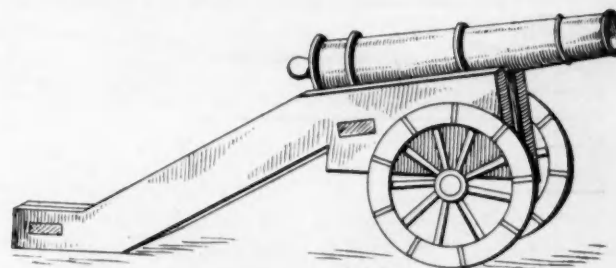


FIG. 3—EARLY TYPE OF WHEELED GUN MOUNT

In Fig. 3 the elevation of the gun was changed by wedges driven between the intermediate transom and the breech end of the gun. This type was superseded by a gun which was mounted on a carriage provided with an arc. A lug on the breech end slid in this arc and the angle of elevation was set by placing pins through holes in the arc and lug. This type was used until the end of the Napoleonic wars.

Fig. 4 shows a simple cast gun with sliding breech block. This was the first gun to use the breech-loading principle successfully. The projectile was inserted

through the breech and fitted into the rifling. The powder was next placed in the chamber and rammed home. A fuse was inserted in the touch hole and the breech

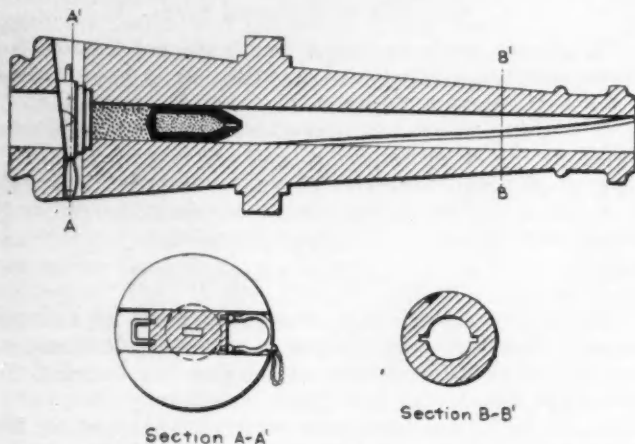


FIG. 4—FIRST GUN USING BREECH-LOADING PRINCIPLE SUCCESSFULLY

closed by means of the copper plug. The wedge was then driven home, forcing the copper plug tightly against its seat so that it would not allow gas to escape. The force of the explosion merely tended to force the wedge backward and not to slide out of its seat.

Following these operations, fire was applied to the fuse sticking from the vent hole—which in turn ignited the powder in the chamber. This was converted into gas and, due to the pressure, the shell was driven forward; the rifling grooves hold two lugs which were cast on the exterior of the shell, causing it to rotate. This rotation was necessary, as otherwise the shell would tumble end over end and its range would be decreased.

Built-up Guns

Fig. 5 shows a built-up gun consisting of a tube A or inner cylinder with a jacket B shrunk over it. The tube

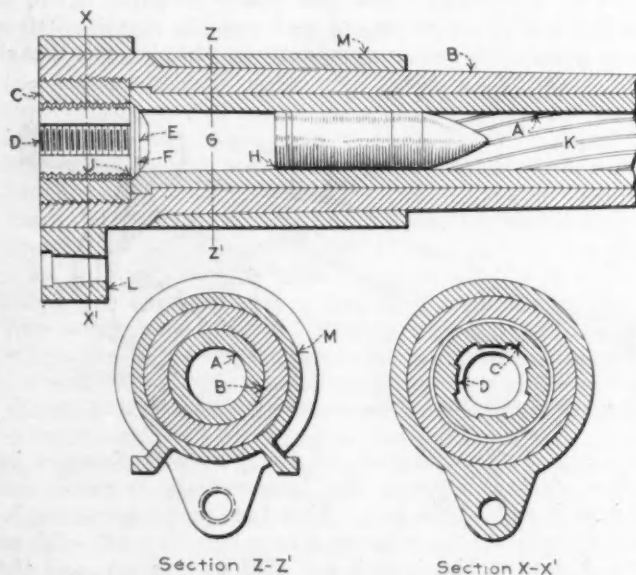


FIG. 5—LONGITUDINAL AND CROSS SECTIONS OF A BUILT-UP GUN

is recessed at the breech end to form the powder chamber G and is closed by the breech block D, which fits into the bushing C as shown. This bushing has an interrupted thread with three blank portions milled out.

The breech block with a similar arrangement of threads and blanks is inserted and locked in position by being rotated through 45 deg. The escape of the powder gases is prevented by the gas check pad F, which rests upon the bevel slope J. By the action of the gases in the powder chamber, the head E is forced back, compressing the gas check pad firmly against its seat. The centering slope H enters the shell and the rifling K gives it the required rotation. The breech ring L is provided with a lug and hole by which the recoil mechanism is attached to the gun. The guide ring M carries the guides, which allow the gun to slide in the cradle while recoiling.

The jacket is a forged steel tube which is bored to a diameter slightly smaller than the exterior of the tube

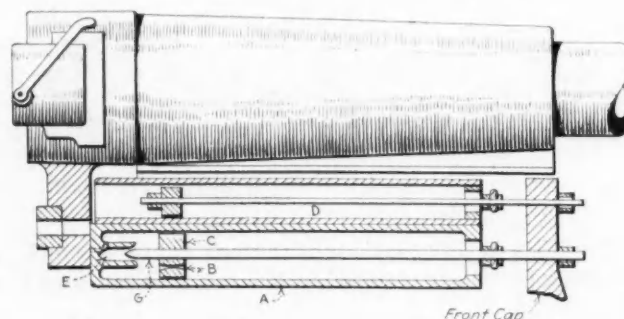


FIG. 6—CROSS SECTION OF TYPICAL RECOIL MECHANISM OR RECUPERATOR

A—Oil cylinder. B—Orifice. C—Brake piston. D—Air cylinder. E—Counter recoil buffer. F—Counter recoil valve. G—Buffer plunger on piston rod

on which it is to be shrunk. Shoulders are counterbored so that the tube will not be driven out of the jacket while firing. Sometimes the jacket is threaded instead of being provided with shoulders, but this complicates the manufacture and shoulders are more universally used. The jacket is heated in an oil or electric furnace until it has expanded sufficiently to slip over the tube. When it has reached the required temperature it is removed from the furnace and placed over the tube. To ensure a contact at the breech shoulder this end of the piece is cooled quickly by spraying with water, which causes it to contract and grip the tube at this point. The rest of the forging is now cooled slowly so that it will contract toward the breech end. When finished, the compression is about 50,000 lb. per sq. in. as a general average.

The breech end of the gun is next threaded so that the breech hoop or ring can be assembled. This hoop or ring carries the breech block. In cases where a plug

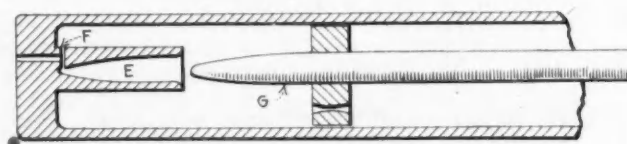


FIG. 7—ENLARGED VIEW OF COUNTER RECOIL BUFFER

is used, the breech hoop is provided with a bearing for the breech block carrier arm, and when the sliding wedge type of block is used the breech ring forms the seat for the block and the bearing for the arm and operating lever.

The guide rings are often omitted and the feet are shaped directly from the jacket. These feet slide in a groove cut in the cradle and guide the gun during recoil.

Recoil-Regulating Devices

There are two main systems of recoil regulating devices: the spring and the hydraulic. In the spring type the energy of the gun is absorbed by the work done in extending a heavy coiled spring and the piece is returned to the firing position by means of the spring. A counter-recoil buffer is provided so that the gun will not fly back violently after it has reached the limit of its recoil. This is usually some type of dash-pot.

In the hydraulic device the energy of the recoiling gun is converted into the work of forcing a liquid, usually oil or glycerine, through a small orifice or past a valve. The counter-recoil is effected by springs or compressed air, a buffer being provided as before to ease the shock of return. Fig. 6 shows the cross-section of a typical recoil mechanism. The cylinder *A* contains oil which is forced through the orifice *B* in the piston. This piston is attached to the frame of the carriage while the cylinders recoil with the gun. The cylinder *D* contains air under an initial pressure sufficient to maintain the gun in the firing position at the maximum elevation. Upon recoiling the air is further compressed, which also creates resistance against which the gun must work during recoil. During counter-recoil the compressed air expends some of its energy in bringing the gun back to the firing position and an appreciable amount in forcing the liquid through the orifice in the piston *C*. A counter-recoil buffer is provided at *E*, Fig. 7, where the extension of the piston rod *G* enters a dash-pot having a minute opening which is regulated by the valve *F*.

winding. By Lamé's formula for hollow cylinders the tangential unit-stress

$$S = r_1^2 R_i \left(1 + \frac{r_2^2}{x^2} \right) \frac{1}{(r_2^2 - r_1^2)}$$

where R_i equals the interior pressure and r_1 and r_2 equal the interior and exterior radii of the cylinder and x is the distance from the axis of the cylinder to the point where the stress is being measured.

By solving this formula we find that the value of S is greatest at the inner radius and is equal to

$$S_i = R_i \frac{(r_2^2 + r_1^2)}{(r_2^2 - r_1^2)}$$

The use of this formula may be shown by giving the results of shrinking a hoop upon a tube so that a pressure of 5000 lb. per sq. in. exists between them. Let the inside and outside diameters of the tube be 4 and 6 in. respectively, and let the interior pressure owing to powder gases be 25,000 lb. per sq. in. By solution of Lamé's formula the resultant tangential unit-stress in the bore is found to be 47,000 lb. per sq. in., as compared to 65,000 lb. per sq. in. which would result if the tube were not hooped. If the same figures are used, except that the unit pressure caused by the hoop is increased to 10,000 lb. per sq. in., the resultant tangential stress would be reduced to 29,000 lb. per sq. in. at the bore.

Wire winding tends to reduce the size of jacket or covering necessary to obtain the same initial unit pressure, for it has been shown by experiment that the pressure resulting from a series of unit pressures is the algebraic sum of the tangential stresses set up by each one individually.

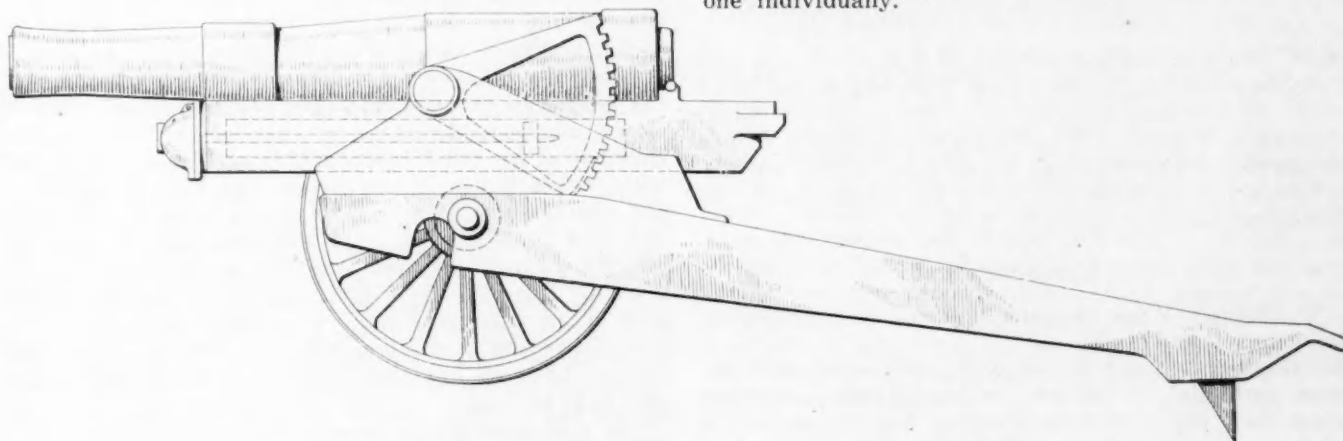


FIG. 8—SHOWING THE COMPONENT PARTS OF A RECOIL TYPE OF GUN

Fig. 8 shows a complete gun with gun proper, recoil device, cradle or sleigh, carriage and trail.

The recoil device is carried in the cradle, which can be elevated or depressed. The lateral traverse of the piece is obtained by the carriage, which is hinged directly to the trail. For deviations greater than those possible with the traversing gear on the carriage, the trail must be moved.

The trail is provided with a spade which prevents it from recoiling and steadies the piece during fire.

Calculation of Gun Stresses

The strength of a gun is obtained in two ways, first by shrinkage as before described and secondly by wire

Two shrinkages, however, should be noted—the absolute and the relative shrinkage. The absolute shrinkage is the difference between the exterior diameter of the tube and the interior diameter of the jacket. The relative shrinkage, on the other hand, is the absolute shrinkage divided by the radius of the contact surface.

The firing of the shell creates a pressure which tends to expand the inner tube. If the pressure is great enough it will break off. If the tube or jacket were not shrunk on, the pressure would break the inner tube. The limit of stress would be the point at which the tube would break during firing. During the process of shrinking on the jacket the inner tube is reduced a certain amount; this compression of the tube adds to its strength. Before it can exert any strain on the tube itself, the force of the

expanding gases must not only overcome the pressure that the jacket places upon the tube, but in addition the resistance of the metal in the tube. The stress imparted by the shrunk jacket, plus the strength of the metal itself, makes it possible to build high-power guns as we build them today.

Referring to Fig. 9, where ab is the absolute shrinkage and R_o and R_i the interior radius and the radius of the surface of contact, the relative shrinkage

$$\Phi = \frac{ab}{R_i} = \frac{oc + ci}{R_i}$$

The relative compression is C/R_i , which is the total compression per unit of radius. It is this value which governs the stress in the tube.

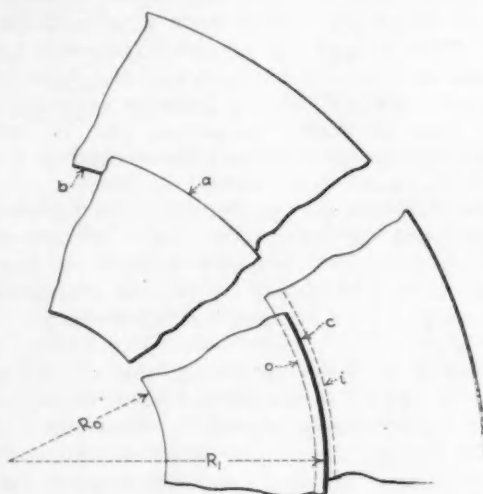


FIG. 9—ILLUSTRATING ABSOLUTE AND RELATIVE SHRINKAGE

Most compressions and shrinkages are given as a certain amount per inch when worked from the design viewpoint, but for shop use the total shrinkage is given. Thus, taking a tube of 6-in. diameter with 0.001 in. per inch shrinkage, the drawings would show a total shrinkage dimension of 6.006 in., while the jacket would be bored to be exactly 6 inches.

GUN CLASSIFICATION

The different types of guns in use may be roughly divided into four classes.

Mortars—These are pieces with very short barrels, seldom more than 8 calibers in length, which operate against their objective with plunging fire. A mortar is used at elevations between 45 deg. and perpendicular. It should be explained that the term caliber, as applied to ordnance work, means the number of times the diameter of the gun is contained in its length. Thus a 12-in. gun, 45 calibers long, would be 45 ft. in length.

Howitzers—These are used for plunging fire and are also guns of relatively low velocity, seldom exceeding 1500 ft. per sec. They are between 8 and 15 calibers in length and operate between the elevation angles of 15 and 45 degrees.

Field Guns—These are guns of high velocity, seldom operated at an angle exceeding 15 deg. and depend upon velocity for the depth of their effective range. They have muzzle velocities up to 2000 ft. per sec. and are from 20 to 25 calibers long.

Rifles—There are naval and siege rifles. These pieces are from 40 to 50 calibers long and employ velocities above 2000 ft. per sec. Their range is very great and

their accuracy of fire much superior to that of field guns. Although they are longer and therefore harder to mount, some large rifles have been used on railway carriages by the French, and the British have also mounted their excess naval guns in this way.

The requirements of field artillery are threefold—that it be mobile, that its fire be accurate and rapid and that repairs can be easily made. In late years mobile artillery has come to include almost every size and type of gun. Howitzers as large as 16 in. have been constructed which can be removed in a few minutes, should the enemy artillery commence to find their range. Some guns as large as 12 in. have been mounted directly on wheel carriages which can be hauled away by tractors. Railway mounts have been developed which will carry almost any gun which can be designed. Almost every type of gun, therefore, except those permanently mounted, may be called mobile. The really mobile artillery, however, includes the light guns—those below 5 in., which are either horse or tractor drawn and can be operated over any kind of ground. The limit of weight in these pieces is usually about 8000 lb. for the gun and limber. The distribution of weight is very important. If there is too much weight on the wheels, the gun will become mired in soft ground; the weight per wheel is usually kept below 2 tons. In the 4.7-in. howitzer the gun itself can be detached from the recoil mechanism and slid out on the trail, helping to distribute the weight. Abroad the piece is sometimes dis-assembled and then carried on separate wagons.

Rapidity of Fire

Rapidity of fire is obtained by the use of a breech block which is light and easily operated, needing only a small angular turn to lock it firmly in place, and by the use of fixed ammunition, with the propelling charge carried in a copper case firmly attached to the base of the projectile. In the latest designs the breech block is closed automatically when a shell is thrown into the chamber of the gun and the empty cartridge case ejected when firing has been completed. As high as 30 shots per minute have been fired from a 75-mm gun in practice, and while this is not a practical way in which to use the gun, it demonstrates the advance of artillery science. Even the great howitzers in use abroad can be fired at intervals well under a minute, which shows the gain above the guns of a few centuries ago, when the damage done by one shot could be repaired before the next shell arrived.

The non-recoiling carriage, which preserves the alignment of the piece, has also helped to increase the rapidity of fire.

Accuracy of Fire

Accuracy of fire has been improved by the refinements introduced in ammunition and fuses and a more complete understanding of the functions of the rifling in the tube. At present the rifling not only gives the shell the necessary rotation that it may maintain its flight, but also arms the fuse.

Operation of Fuses

Looking at the sketch of a fuse, Fig. 10, we will understand how this is accomplished. The left-hand section shows the firing point below the collar of the striking weight. Should the shell be dropped or struck when the pin is in this position the fuse would not ignite, but if the pin were above the collar as shown in the central

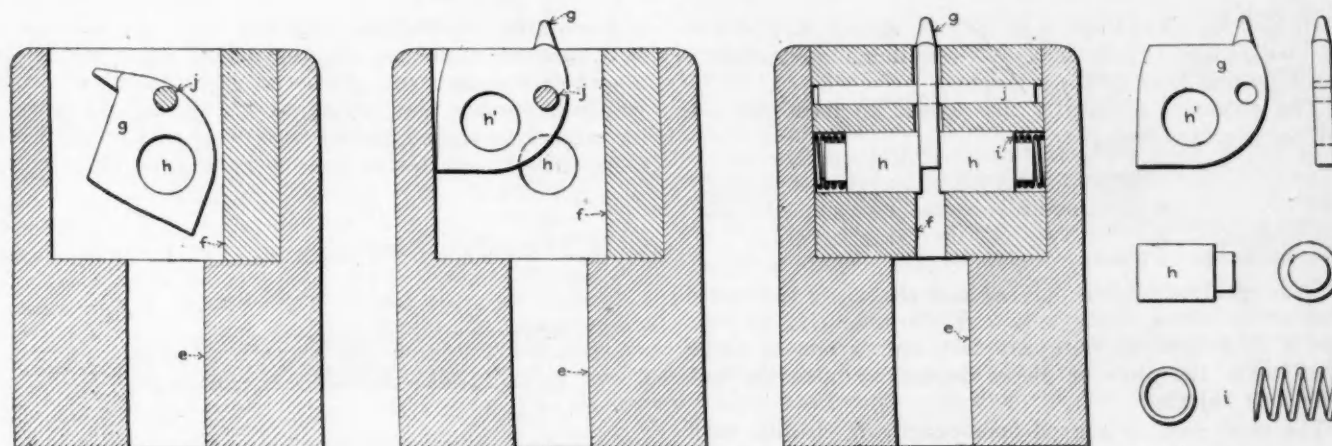


FIG. 10—ILLUSTRATING THE OPERATION OF FUSES

section the firing point would be forced into a detonator, which would ignite the powder train and explode the shell. When the shell rotates the two lugs shown in the right-hand section tend to move outward, but until the centrifugal force can overcome the resistance of the coiled springs this cannot occur. When the force becomes great enough the lugs move outward, allowing the pin to swing clear and the firing point is raised into the "armed" position. A similar principle is employed in many types of fuses.

Referring to Fig. 10 it will be noticed that *hh* are the two lugs referred to above. There is also a hole in the firing point, corresponding to lugs *hh* (see *h* in left-hand view and *h'* in right-hand view). Normally when these lugs are locked, the shell in revolving throws the pins out, allowing the point to come back into firing position so that if the shell should hit the ground or other obstruction, the detonator or firing cap would be punctured by that firing point and this might explode the shell. This percussion device is used today in the explosive shell.

Size Limitations of Guns

I have been asked if there is any limit to the size of guns. Theoretically, there seems to be none, but the practical use of the guns automatically sets certain limits. The first limitation is fire control. Unless one is able to observe the effect of the fire and correct any errors, the possession of guns which out-range those of the enemy gives no advantage. The airplane has made observation of artillery fire possible beyond the range of any dream. Railway depots and ammunition dumps miles behind the front lines have been shelled, while hostile aircraft signal the results to their batteries by wireless, corrections being made accordingly. The result of such work was mentioned earlier in this paper, but I will repeat it to illustrate my point. On one section of the front the British wished to destroy a freight depot where large masses of munitions were apparently being concentrated. This was more than 10 miles behind the German line and the British guns were 17 miles from their objective. A large gun of about 12 in. diameter was used. Two ranging shots were fired, the second being a direct hit which struck one end of the depot. Five other shots were fired and when the smoke cleared away the aviator who had directed the fire of the guns took a picture of the result. Six shell craters, roughly 50 yd. across, were all that remained of this station. If we increase the range, we find that the dispersion of the shots increases rapidly and owing to atmospheric

irregularities the probability of accurate fire is reduced. Furthermore, the airplane cannot signal, as its wireless is not strong enough, and therefore the gun must be fired blindly.

The other limitation is the cost of such super-cannon. A large gun has, at best, a life of a few hundred rounds and only a small number of these are accurate. The gases corrode the bore much more quickly in large guns than in the smaller sizes, owing, no doubt, to the great amount of powder burned and velocities attained. The great rifles with which the Germans are shelling Paris must have an enormous powder chamber to maintain pressure sufficient to allow the velocity of practically 4000 ft. per sec. to be reached. This is almost double the velocity of our field guns. It must be nearly 80 calibers long to give the required velocity without exceeding the elastic limit of its material by too large a margin. It is therefore probable that this gun is set upon a permanent foundation and cannot be shifted about. Naturally an objective the size of the city of Paris is easy to hit if the range can be had, for if the shell lands anywhere within the city the Hun will have attained his desire. Such a cannon is expensive to build and this one must have cost a staggering figure. Only the Germans, with their belief that terror will win, could have converted their money into such a cannon. It is far better for us that they would rather spend a quarter or a half million dollars on one gun the life of which can be measured by days than to spend the same amount for a large number of field pieces that could put up a barrage fire before our trenches.

GUN INSPECTION

The inspection of artillery material is very rigid. We are bound by wise rules in order that the army and navy may be supplied with the best material and that it will be uniform. Long years of experience have shown that certain requirements must be met or the men in the field will suffer; rules which are strict but just are enforced. Some persons have complained that this rigid inspection of material costs too much money. In answer to that, is it not better to pay a few dollars more to know that a gun is right and safe than to save those dollars and perhaps spend human life in its stead?

Chemical and Physical Requirements of Gun Metal

Our regulations lay down certain requirements for forgings for guns which can be briefly summed up as follows: The chemical analysis is approximately carbon

0.35 to 0.50, manganese 0.50 to 0.80, silicon 0.15 to 0.30 and nickel 2.50 to 3.50 per cent. Sulphur and phosphorus must be less than 0.05 per cent.

The chemical properties are varied to meet the following physical tests:

	Elastic Limit	Tensile Strength	Elongation	Contraction of area
	Lb. per sq. in.	Lb. per sq. in.	Per cent	Per cent
Tubes and jackets...	65,000	95,000	18	30
Small parts and breech blocks...	75,000	110,000	14	30

Heat treatment by tempering and annealing is allowed and three official tests only are allowed. If the test pieces submitted do not pass test during one of these three tests, the piece or pieces depending upon the test piece are rejected.

The steel used is a good open-hearth or electric steel free from slag and bottom, cast in either open molds or special ingot-casting machines. The specifications demand that the cross-sectional area of the ingot as cast be at least four times the area of the block forging which will be made from it. If bored or punched ingots are used for hollow forgings, the walls must be reduced 50 per cent under the hammer. The lower end of the ingot will always be the breech end, as it is usually more free from defects than the top.

A discard from the ingot of at least 30 per cent by weight from the top and 5 per cent from the bottom is required in all open cast ingots and is slightly less for fluid compressed ingots.

Forgings are made on the hydraulic press, steam hammer or on the drop hammer. The use of the press is usually confined to forging shields and other large plates, the hammer is used for blocks, tubes, jackets, rings, etc., while small parts and difficult sections are made by drop forging.

HEAT TREATMENT

Forgings are annealed above their upper critical point after forging and are then rough machined. After being rough machined they are heat treated so that the steel will be given the necessary physical properties as called for in the specifications. In heat treating the whole piece, never but a part, should be subjected to the same treatment, and this also applies to the quenching operation. Likewise, in annealing, the whole piece should be drawn as evenly as possible and where possible the forging and furnace should be allowed to cool to not more than 200 deg. Fahr. before removing it from the furnace. If this cannot be done the furnace and the piece must be allowed to cool below 600 deg. Fahr. before it is removed and then allowed to cool slowly in air. The last operation in heat treating a piece must always be a drawing or annealing treatment.

No treatment is allowed after the piece has passed test.

Test pieces are usually taken from both breech and muzzle end of the forging, and tangential bars are taken whenever possible. This is because the tangential unit stress is always the limiting feature of gun design, the longitudinal unit stress being much less important.

Standard Test-Specimens

The standard test-specimen is a bar 4.75 in. long, 0.75 in. diameter, threaded at both ends for a distance of 1 in. with a $\frac{3}{4}$ -14 U. S. Standard V thread. The center position between threads is turned to 0.505 in. diameter for a distance of 2.2 in., filleting into the threaded portion with a 0.375 radius. The 0.505-in. diameter gives an area of 0.2 sq. in.

Machining of finished material must be carried on with utmost care and all dimensions kept within the allowable tolerance as shown on government drawings which accompany the specifications covering the particular material. Rough and careless work and finished surfaces showing chatter or tool marks will not be accepted.

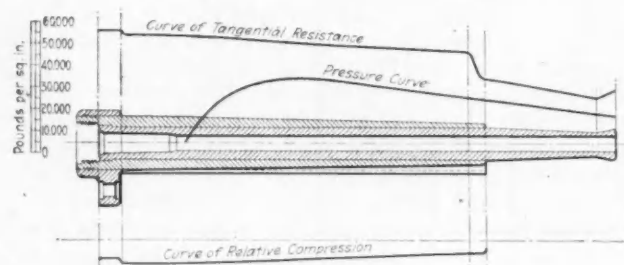


FIG. 11—CURVES OF GUN PRESSURES AND TENSILE STRENGTH

Where exceptional finish is required, file and emery polishing are usually specified. Grinding is allowed but surfaces are inspected for seams and cracks before grinding, as that operation usually peens out such defects. Cold "shuts," seams, struts, blow-holes, sand or slag pockets, pipes or any other structural defects are looked for at each inspection and are cause for rejection. Even "ghost lines" of oxide are carefully investigated and if numerous or in dangerous positions, will cause the scrapping of the forging.

Gun Factors of Safety

The importance of such rigid inspection can be seen from Fig. 11, which gives the tangential resistance, pressure and velocity curves for a hypothetical gun. As can be seen, the elastic limit of the forging is nearly approached by the compression due to the jacket. The pressure curve shows how the gun is strained during firing. At one point, the value of the tangential resistance is 50,000 lb. per sq. in., while the stress produced by the interior pressure is 35,000 lb. per sq. in. This gives a factor of safety of 1.43, which is low compared with engineering standards, where 5 is a common figure. We will assume that this compressive strength was obtained by 0.006 in. shrinkage. It is easy to figure the effect that a difference of 0.001 in. above or below the 0.006 in. would have. Say the absolute shrinkage had been 0.005 in., then the value of the resistance would be reduced to about 40,000 lb. per sq. in. and the safety factor to 1.14. If the safety factor is reduced below 1, the tube will explode. On the other hand, assuming that the shrinkage is 0.007 in., the safety factor in the tube will be increased, but the metal of the tube will be stressed above its elastic limit. This, in itself, is not dangerous as the walls are well supported, but the danger lies in producing too much strain in the jacket when the internal pressure of the tube is communicated to the outer hoop, with the likelihood of its being split off when the gun is fired.

Is it surprising that when working with such small safety margins we are bound to reject any piece suspected of being weak in the slightest degree? A minute crack in a jacket, too small for the eye to notice in a hurried inspection, may indicate a seam beneath the surface, and is it not better to waste work already done on a piece rather than run the risk of needlessly sacrificing a human life?

Fig. 11 is a cross section through the axis of the gun. Now, when the jacket is shrunk on there will be compression in the inner tube. This is only a theoretical drawing, the tangential unit strength being assumed as 50,000 lb. per sq. in. This strength would drop suddenly at the end of the jacket, and the unit stress would be just about the stress of the tube beyond the end of the jacket. The pressure starts at the powder chamber and runs up to the maximum. As the shell travels along, the burning powder cannot keep up that maximum pressure, and it decreases. A velocity curve would show an increase as the shell travels along the bore of the gun. This, we will say, was made with 0.006 in. shrinkage. This means that the outer diameter of the tube was 0.006 in. more than the diameter of the jacket. They were put together by heating the jacket until it was large enough to fit over the tube. With only 0.006 in. shrinkage, which the inspector passed, instead of getting a pressure of 50,000 lb. per sq. in. in the gun, it will be 35,000 lb. As I assumed in my notes, the shrinkage is only 0.005 in., which reduces the compression and therefore the strength in the tube, at the point where the jacket touches the tube, to 40,000 lb. Instead of having a factor of safety of 50,000/35,000 we will have 40,000/35,000, or 1.14. Thus it will be seen that one or even a half-thousandth will make a great deal of difference. On the other hand, if the shrinkage is made 0.007 per in. we will get 65,000 lb. per sq. in. and exceed the elastic limit of the tube. While that will not do much harm it will put too great a stress on the jacket, and when the gun is fired a stress will be introduced in the jacket and it may split off.

Engineering Problem of the War

We must have guns and guns and more guns. We know that guns are the only weapons with which we can kill the most Huns at the least cost to our own army. Every step in ordnance work must be studied and simplified and speeded up. We are today boring and turning the tubes and jackets at a rate which was believed impossible a few years ago. We are turning out shells by the hundreds from machines that were turning out but dozens before the advent of "stellite" and other cutting metals.

We will cite an example which has lately come to notice of the way that engineering methods have speeded up our work. The rifling of guns has always been a long and tedious operation. A rifling machine consists of a cutting tool carried on a bar which is made to rotate at the required rate while moving longitudinally through the bore of the gun. The tool thereby cuts a spiral groove in the bore of the gun, the shape of the groove being similar to that of the cutter used. One, two or four grooves have been cut simultaneously, but when more than one set of grooves is cut, difficulty is experienced in keeping their depth uniform. Lately this method has been superseded in a certain gun shop by a set of broaches which has cut the time of rifling from 6 hours to 45 minutes for the particular type of gun made there.

Hundreds of other problems in ordnance work will show similar savings of time if they are given the attention which they deserve. Casting, forging and machining problems may be cited, and there are sheet metal problems in the carriage and limber work and difficulties in all branches which can be solved by special jig, fixture or tool work if careful study is devoted to them by our best engineers.

For success armies are dependent upon their equip-

ment, and they cannot gain victories unless they have a superiority of material. The engineers of this country must see that they get the material, not only in the best possible condition for effective use, but in sufficient quantities.

There is no gain in having the best guns in the world or the best planes or trucks, if we have only a few, for the Germans have turned their whole attention to making these articles and are getting out vast quantities of them daily. Unless we have more of them than the Hun, we shall not be able to defeat him quickly and we will have to pay dearly in lives and money, but if we unite and give the problem of smashing his power our full and undivided attention, we will most surely be able to redeem France and Belgium and make the world safe for democracy.

DISCUSSION

A MEMBER:—How accurate must the tripod mounting of the machine guns be made? I heard the remark not long ago that a machine gun, like a lawn sprinkler, does not require very great accuracy in the parts of its mounting. I refer not so much to the machining as to the clearance of rotating parts and traversing arc, and end play in the elevating screw.

LIEUT. SKERRETT:—The specifications for parts of the gun must be lived up to, and they must be machined accurately, so that when put together there will be no shake and chattering, or vibration due to looseness in the mounting.

THE MEMBER:—The material in the inner tube, initially in compression, as I understand it, is in tension every time the gun is fired, and then comes back again. Would this not cause fatigue of the metal?

LIEUT. SKERRETT:—I do not believe the reversing from compression to tension and back again occurs enough in the ordinary life of a gun to give much trouble. It may in some of the lighter field pieces, but they are not usually compressed to a dangerous extent, nor would the reversion produce crystallization or failure of the material.

CHAIRMAN HINKLEY:—Is the tube in compression when at rest?

LIEUT. SKERRETT:—It is in compression when at rest, and in firing it is in tension. The jacket is always in tension.

CREUSOT 75-MILLIMETER GUN

N. C. BANKS (A.S.A.E.):—We have heard a good deal about the Creusot gun and how successful it was at the beginning of the war. It is still in use. I am told its success is due to the recoil mechanism. Why have the Germans never been able to imitate that? I asked in Paris once about the secret of this gun's recoil and was told that it was one of the things they could not show me. A number of 75-mm Creusot guns have been captured, we know, and why are the Germans not able to imitate them so as to obtain accurate firing while taking up the recoil?

LIEUT. SKERRETT:—The French seem to be able to destroy those guns before they are captured and thus keep the secret. On all guns (especially the hydro-pneumatic guns, which employ a liquid and compressed air for returning the gun into firing position after it has been discharged) there is a way of determining the size of the orifice through which the liquid is forced, and also of obtaining a proper recoil buffer. Those two points are the hardest to imitate and the formula by which they are obtained is hard to work out.

The pressure developed in retarding the gun is what

slows the gun up and then brings it back into its firing position. The design of this orifice is very complicated; a sort of butterfly valve, fitted into the rifling of the cylinder, rotates under the guidance of the rifled grooves and in so doing increases the size of the orifice, or, as in the case of the "75," a throttling bar is used. (See Fig. 6 and the description which follows.) The French have worked over the design of this orifice ever since 1870, but they did not perfect it until about 1898 or 1899. They spent twenty years designing just one gun. The Germans did not have an opportunity to get hold of a gun of this type until 1914, and they could scarcely succeed in designing one which would equal it in three or four years.

MR. BANKS:—They claim they had succeeded in getting the design before the war broke out.

LIEUT. SKERRETT:—Well, their guns do not show it.

C. F. JEFFRIES (M.S.A.E.):—Only a few months ago I saw in the magazine, *Artillery*, a picture of German artillery and officers stationed on the German eastern front. It was a picture of action behind the German lines, and in it were French "75" guns. The picture may have been a fake, but it showed the same gun down to the most minute detail that could be seen in a picture—the same construction of breech, and nothing to show that there was any difference at all.

LIEUT. SKERRETT:—They may have been French guns, but I doubt it. They might have been, like the tractors, sent over in catalogues. The Germans have used a design of the same type for the breech block of their field guns for twenty years. The breech is offset, with the load hole below, as in the French gun. The bore of the gun is above and the breech block is slid around and the shell inserted; then the breech block is brought, through a half turn, solid against the back of the shell.

A MEMBER:—How many rounds is the life of ordinary field artillery? It is my understanding that the rifling of the guns gives out, and that they are then scrapped. Could not shells slightly oversize be used?

LIEUT. SKERRETT:—After a certain number of rounds, it varies with the gun and the conditions under which it is fired, the guns are usually rebored, a liner is inserted and the guns rerifled. They are not allowed to make oversize shells.

A MEMBER:—The paper refers to an accuracy of a thousandth of an inch in boring. In turning a 6-in. shrink surface what method is used to maintain that accuracy? Is the barrel ground on the outside, or simply turned smooth?

LIEUT. SKERRETT:—The usual practice is to grind it. We do not expect the jacket to come to us bored to a thousandth of an inch all the way through. As a matter of fact, we get a jacket supposed we will say to be 6 in., but it will be 5.992 in. one place and further down 5.991, and in another spot 5.989 in. Now we have to put a 0.006-in. shrinkage on that. We lay out a tube drawing; the dimension at the first point would be 5.992 + 0.006, that is, 5.998 in. At the second point it will change to 5.997 and at the third point it would be 5.995 inches. With such a drawing we can turn a tube which will fit the jacket, because it is always easier to turn an outside surface than an inside one.

A MEMBER:—I should think there would be considerable variation in a long tube.

LIEUT. SKERRETT:—There is, but in boring guns we get good results, because very little is taken out in the last boring, sometimes only two or three hundredths of an inch.

The guns are reamed with a packed bit whose cutting edges are sprayed with oil and are micrometered to an exact size. The reaming bits are entirely lined with wood, just to diameter. The first one takes out $\frac{3}{8}$ in., the second rougher takes out only a couple of hundredths of an inch.

H. M. JEROME (M.S.A.E.):—Lieutenant Skerrett has used a wider range of limits in his illustration of jacket-bore dimensions than would be found in actual practice.

A MEMBER:—I once visited a gun shop and saw them taking those measurements every inch. The jacket diameter was some 30 or 40 in., and I do not know how many reamers they ran through it. The amazing part of this micrometering is the accuracy with which the old workmen can take the measurements in those big, long holes. I do not know just how they do that today, but then what is known as a star gage was used. It is a micrometer gage about a yard long which is placed in the bore but is read on the end.

A MEMBER:—Does not the scale produced in heating the jacket have some effect?

LIEUT. SKERRETT:—The heat is only moderate, very seldom above 750 degrees. Scale will not form at that temperature.

A MEMBER:—What method is used in attaching the insert or liner to a worn gun?

LIEUT. SKERRETT:—We rebores the tube. Sometimes, if it is a small gun, we bore out the tube entirely. The jacket goes over the tube, as in the original gun. This is pinned in with a lock or key, so that it will not revolve when the shell starts through it.

VERNE JACKSON:—How is the liner put in?

LIEUT. SKERRETT:—It is shrunk in; the entire gun is heated.

J. G. SMITH:—What is the method of shrinking the jackets on. Of course they are heated, but how are they handled? Are they swung on horizontally?

LIEUT. SKERRETT:—They are put on in a vertical position usually; the tube is stood upright and the jacket dropped over it. Sometimes the jacket is held and the tube is dropped into it.

This depends on which method is preferred. The jacket is then cooled at the breach end and shrinks around the tube compressing it. We always try to get a grip at the breech shoulder of the tube and draw the rest of the jacket that way as it cools.

Long Range Guns

GEORGE AINSWORTH:—Have American artillery officers ever developed a gun to compare with the long-range German gun.

LIEUT. SKERRETT:—That I do not know. Back in 1875, Ingalls at Fortress Monroe, who was the artillery expert of the army at that time, stated that it would not be hard to develop smooth-bore guns with a 50-mile range. I think it would be possible to develop them, but I do not know that we ever have.

C. P. GEEN:—In wire-wound guns, which are still used considerably, is the tube in compression, and if so, how is it put in compression; in winding the gun?

LIEUT. SKERRETT:—It is just like wrapping a rubber band around one's finger. First one thin jacket and then another is applied. In other words, the compression is cumulative.

Stream-Line Shells and Vacuum Effect

MR. GEEN:—Has any serious thought been given to de-

creasing the vacuum or eddy effect behind the flat back of shells? This is considered to have a great retarding effect.

LIEUT. SKERRETT:—The high-powered guns we are using now create a vacuum behind the back of the shell, but I do not know the amount of the actual retardation. Whitworth, in 1866, had a curiously shaped shell for a gun with a twisted hexagonal bore, one turn in twenty-five. The shell looked like a loaf of bread, and it acted in flight just as a loaf of bread would. I do not know that a shell with stream lines would have a considerable effect on the range, nor do I know that it has ever been tried.

MR. GEEN:—I understand that one that was tried had a detachable piece on the end, which flew out from the projectile after a comparatively short distance. That piece had a flat back and was bored to fit the projectile, but the projectile, I believe, was heavier at the front and followed an oscillatory motion.

LIEUT. SKERRETT:—The only objection to giving it a stream-line form is that the driving band is so far forward. When we place a shell in the bore of a cannon we want to have as good a surface as possible for the gases to act upon and in order to prevent all losses around the edge of the shell; we are dealing with pressures up to 35,000 to 40,000 lb. per sq. in., and they are hard to hold.

C. C. HOPE:—All powder cases have thus far been made of brass or copper. Is there any reason why steel could not be used?

LIEUT. SKERRETT:—The brass case acts as a seal for the powder chamber. A steel case will work all right at first, while a gun is new, but when there is a clearance of a thousandth of an inch the steel will not expand enough to hold tightly to the walls all around. There will be leakage and this will cut down the efficiency of the gun a great deal. Besides this, steel will not stand the pressure of the discharge of a gun, as it is liable to fracture.

Rifling Methods

R. G. HANDY (Jun.S.A.E.):—How is the broaching done, and how deep is the rifling?

LIEUT. SKERRETT:—Broaching has not been tried on these larger guns as yet. It has been tried with small guns only. Up to this time we have cut one or two grooves at once or even four, by changing the position of the head-stock from point to point as the grooves are put in. What we do nowadays is to put on a cutter and outline the rifling all the way around. It can be done in one operation. Broaches that take anywhere from 0.002 to 0.003 in. at a cut are run through. The rifling is about 0.02 in. deep.

MR. HANDY:—What is the relation between the diameter of the bands on the shell and the bore of the gun?

LIEUT. SKERRETT:—It is slightly larger than the bore of the gun, to insure that the copper is driven tightly into the rifle grooves. This soft copper driving band is forced down into the groove as the shell is driven forward.

C. G. COWLES (M.S.A.E.):—What is the clearance between the shell and the bore?

LIEUT. SKERRETT:—It is usually 0.01 inch or less. However, some of them are more.

Double-Acting Shells Not Practicable

MR. COWLES:—One of the main stresses in firing is

the longitudinal stress which has to be counteracted by means of the inertia of the gun carriage and the gun, plus the resistance of the recoil mechanism. As I understand it, that is difficult to do in such frail carriages as airplanes. Furthermore, it is difficult to bomb such objectives as railway stations, owing to inability to aim accurately. The shell should be shot and not dropped in order to obtain the accuracy desired. Why is it not possible, instead of supplying an airplane with a mechanism that may tip it over or perhaps destroy it, to design a double-acting shell, so that, while one shot would be wasted in going in the opposite direction, still it would supply the force which would do away with the kick on the machine?

LIEUT. SKERRETT:—The French designed and tried out a "one-pounder" acting on just about that principle. It had a double barrel, a one-pounder at one end, and the other was more or less like a choke-bore shotgun. A double-ended cartridge with a side percussion cap was fed into it. This took an ordinary one-pound high explosive shell and a cartridge loaded with buckshot and very close wadding. Action and reaction being equal and opposite, theoretically there would be no recoil with this gun, which was mounted on the frame of an airplane. Unfortunately it did not work out. The first time it was tried the choke-bore jammed and the result was that the full charge, supposed to blow this out at a reasonable velocity and also to act in the opposite direction, all went one way, and the gun blew off the other way with most of the airplane.

One may say that to overcome this trouble it would be necessary to use exactly the same charge and type of shell on each end. The trouble with this plan is that it would be liable to shoot back into our own line.

Field for Automatic Machinery

MR. COWLES:—The author states that every step in ordnance work must be studied, simplified and speeded up. I am sure all of us agree with that statement and I suggest that this can be done by extending the use of automatic machinery into newer fields. I believe we could go into almost any shop and get much more speed out of perhaps one or more operations, just by introducing automatic feeds. For instance, I saw one shop where men were bending sticks for airplanes. About 200 of those sticks are required for one machine and one man feeds and bends about 400 an hour. I believe it is possible to get 180,000 a day per man with perhaps less effort on his part. We are all familiar with automatic screw machine work. For many years knitting machines have been doing most intricate work automatically.

CHAIRMAN HINKLEY:—This war business is new. There are as many automatic devices used in straight commercial work as in the knitting mills. If war were business we would be tooled up for it on an automatic basis inside of three years, without a doubt.

Now, there is always the question as to whether we will get tooled up ahead in time to use our improvements. In the meanwhile, we are trying to increase our mechanical efficiency with such tools as we have. Trench hat making is almost automatic in all its phases. If this were a business that would go on indefinitely, no doubt we would be properly tooled up for it, but the expenditure must be limited by the size of the order. A firm might get an order for three or four million trench hats, yet be out of business in about six months with all the machinery on hand.

Serious Faults of Wire-Wound Gun

A MEMBER:—Why should the Government discard the wire-wound rapid-fire gun?

LIEUT. SKERRETT:—A long gun is usually supported in the middle. It has a bending moment that tends to make the gun droop, and the middle section must be built up strongly enough to prevent this. The wrapped gun is rigid enough at first but the layers start to working over each other every time the gun is fired, owing to expansion of the metal, the droop develops and finally the end gets blown off.

Difficulty of Firing at Airplanes

A MEMBER:—Why is it that a field gun will shoot four or five miles, yet when aiming at a plane 16,000 or 17,000 ft. high it is not as effective?

LIEUT. SKERRETT:—Shrapnel is ranged for 22 seconds' flight. That is only at the 15-deg. angle, the ordinary angle we have always used. If we went up to 45 deg. it is more than likely that we could double the range at the same velocity. Shooting at 15 deg. will give, say, 22 seconds' flight. If we are shooting at an airplane at the maximum range of the gun, the aviator, seeing the flash, can turn and be a mile away before the shell gets there.

We must know how much of a lead to give because those fighting planes fly at 130 to 140 m.p.h., and even in 22 seconds they are a mile away. We must aim a mile

ahead to have any chance of hitting one, and we must count upon its flying straight.

ERNEST GOLDBERGER:—At what angle does the gun shoot most effectively?

LIEUT. SKERRETT:—At about 43 degrees.

A MEMBER:—Can a plane fly higher than we can shoot, or can the gun shoot higher? Do anti-aircraft guns need to be stronger because they shoot vertically?

LIEUT. SKERRETT:—It can outrange the planes, but the chance of hitting them is small. It may be that the anti-aircraft gun is stronger than the field gun. It does not necessarily need a greater powder charge for the vertical work, if they use the same velocity as a field gun.

Armor-Piercing Projectile's Action

MR. AINSWORTH:—What is the difference in construction between the armor-piercing projectile and the high-explosive shell?

LIEUT. SKERRETT:—The armor-piercing projectile and the high explosive shell have different shapes. The latter is filled with high explosives and detonates against the side of the armor. The armor-piercing projectile has a great mass of metal and besides it has a cap of soft iron. When that hits the plate it keeps the shell from splintering at the nose. The projectile punches its way right through and will not explode until it is through. The exterior covering not only gives it a "bite," but acts as a lubricant.

Man Power of the Future

By HARRY TIPPER* (*Member of the Society*)

PENNSYLVANIA SECTION ADDRESS

THERE is no labor problem, but there is a man problem. So long as we deal with it as an abstract mass matter, we shall not approach the solution of the problem in any way. It is not a problem which has just been opened to us by the war. It has been existent for a century, beginning with the transfer from the individualistic craftsman's organization into the mass organization of industry. We have learned through the centuries up to the nineteenth how to lay down a code of laws for the individual, to deal individually in industry, to deal as man to man, but we are only beginning to learn something of the responsibility of mass to mass in industrial operations. It is impossible to study the labor problem of to-day without understanding the history of the labor problem of yesterday. The war has developed the labor problem; it has also developed the labor ideal, and it has brought the question of the ideal of labor and the ideal of the government of industry much closer to us, and has made it much more necessary of solution.

We have seen in this war the force which is given to an applied ideal. No matter whether the ideal be a right or a wrong one, if it is sufficiently accumulated by the mass of people it will in the course of time acquire an operating force and begin to roll. Even the terrible ideals of the German people have, in time, acquired such a unity and operating force that they have been bound to roll in a certain direction. The ideal of labor has been the ideal of the labor union. The only ideal for the mass labor.

The ideal of the individual craftsman was the ideal of accomplishment—the job. The ideal of industry has been the profit. Between the ideal of labor as to hours and remuneration, and the ideal of industry as to profit, there is no common ground or meeting place and there never has been.

From the very beginning of the transfer of industry to the factory system, industry has always exploited labor to the maximum of its capacity, and labor has always gained what it acquired by force. These two things must be thoroughly understood in order to approach this problem. So long as the profits of industry and the profits of labor are the only ideals of industry on the one hand and of labor on the other, so long must the question of difference be solved by strife only; for, unless we displace the ideal of the present with an ideal that can be used in the future for better purpose—unless we get into our work the spirit of accomplishment on both sides, instead of the spirit of profits, we shall continue with the strife that marks present relations. And as this has been brought nearer by the war, it is evident that the labor problem must be met during the war.

Charles M. Schwab said recently that the worker would rule before very long. A few years ago this statement would have been stamped as the rankest Socialism.

The Standard Oil Company, one of the most autocratic organizations in this country, but at the same time one of the most intelligent, began the other day an experiment in democracy in industry. It has arranged that representatives of the workers shall sit with the board

*BUSINESS MANAGER OF AUTOMOBILE INDUSTRIES.

of directors to consider wages, factory conditions and other essentialities of the working men.

The platform of the British Labor Party as outlined by it about three months ago is one of the clearest, shrewdest and most practical documents ever issued by a political party of any kind. No matter whether we agree or disagree with its provisions, we must admire the strength of its language and character of its work.

In England it has been necessary to put committees representing both labor and capital in charge of the war industries controlled by the Government.

In this war we have made a living thing out of the abstract ideal of democracy. We have made it a living thing not in the minds of a few people but in the minds of millions of people. It will be made such a living thing that millions of people will work with no prospect of advancement in money, no prospect of profit, no other incentive than democracy. When that ideal, resuscitated in such a manner, comes upon us after the war, what effect will it have on our industrial relations? How are we going to treat labor as an abstract question, when the ideal of democracy, the ideal that has been expressed through this very sacrifice of war, comes up as a question of industrial relations—the big problem of the waking hours of the populations. Are we to say that labor will be so plentiful that we can go back to the old scheme of industrial transactions and exploit the worker as we have heretofore done? Are we to say that he who has fought for a voice in the affairs of government shall have no voice in the affairs of industry? Is it possible for a government to live which is democratic politically and autocratic industrially, when it has been impossible for two governments to live side by side when one was democratic politically and the other autocratic politically?

FORCE OF LABOR IDEALISM

Shall we deal with this purely as an economic question, or are we alive to the fact that the force of idealism in the labor movement has accumulated such growth in the century of factory work that it is almost ready to roll on? Or, are we simply to sit back and say it will right itself; that we cannot have any interference with our industry; it is our private property; it is our work; we must adjust it as we like in our particular field?

In a town that was seventy-five years behind the times I had the privilege as a boy of watching the transfer of hand work into industry. I saw workers who had contracted for the cloth to be woven in their own homes, in their cottages, transferred to the mill. I saw them get the same pay in the mill as they made from their contract work, but in addition to their pay in the contract work they had been able to weave the cloth for the family. They had been able to attend to their gardens, and while their hours had been long, working from light until dark, they had not been obliged to sit at the machine all the time. Whenever their backs hurt they could get up and recuperate. When they were transferred into the mill they got no more pay, they could not make cloth for their families and could not get the expense out of their gardens. I saw whole families transferred into the mill in order to meet expenses. The individuality of the workman was lost; the individual design and construction of the cloth by the contract worker was lost; he became nothing but a machine operator with two ideas in his head, how to decrease the hours and how to increase the pay. That has been the labor union ideal, and we have constructed no other ideal. Until ten or fifteen years ago we did not try to raise a company ideal

or a department ideal or an organization ideal.

The war has shown us that an ideal is a great operating force; that the incentive is greater than anything else; that industry, just as well as politics, requires ideals, and that politics and industry must accommodate themselves to the same range of ideals.

It is not by chance that the complete autocracy, Germany, has had a greater unity than any other country; the system was the same industrially, politically and socially. In all other countries there has been a division between the industrial and the governmental systems.

The man who has power in politics has no power in industry. He has gained his political power through the growth of democracy. Unless we watch, compromise, understand and perhaps concede, he is likely to demand the Socialistic state toward which he has been tending—is likely to use his political power to demand industrial influence.

I had a letter from a prominent man in London the other day, and he wrote, "There is no doubt in our minds that the reconstruction of England after the war will be in the hands of the Labor Party." Such is the strength of the Labor Party in England. We have not got to that point here, but the same forces are at work here as there, and all depends on the vision of the manufacturers—the vision of the industrial engineer, for the engineer, after all, should have vision as to the rights, obligations and possibilities of the individual man on the labor question, as to whether we enlarge our vision by intelligent compromise, or destroy our civilization by unintelligent strife between the two.

It may seem that I am talking rather wildly here, but anyone can learn how deeply this thing has gone from the documents prepared by officials and committees of the British government. Go into all their work in connection with the hours of labor, the efficiency of the union organizations, the character or the tendency in all industrial relations over there, and finally study the platform of the labor party. Talk to the conservative, intelligent labor leaders over here. They will admit that they have all they can do to keep the radical labor leaders in line at this present moment; that only the unity of the supreme necessity of war keeps the labor unions conservative, and when that necessity has passed and is no longer a force, then the whole radical element of the labor party in some way or other will accentuate its demands with more power, force and energy, because of the recrudescence of these American ideals.

In a time like this, when we have reasonable unity in labor and labor is scarce, when we are troubled with the problem of getting men, when we are competing with each other, and when we do not know what future conditions will be—this is the time when we should study labor as we have never studied it before. If we had put half the study on labor that we have put on a question of engineering equipment, we should be much further along than we are to-day; but there are few men in business who have studied the history of the labor union—its growth, its necessities, the character of its organization, and its possibilities for mass action in the future and the way in which that mass action can be used to better instead of to destroy industry.

The subject cannot be taken up merely from the point of view of economics. It is not economic. Its principles are philosophic, its processes psychologic, and its results economic. If we deal only with economics we cannot solve it. Human understanding must analyze it and human sympathy enter into its consideration, if we are to approach a solution.

Possibilities of the Hvid Engine

By E. B. BLAKELY* (*Member of the Society*)

MID-WEST SECTION PAPER

Illustrated with PHOTOGRAPH AND CHARTS

APPARENTLY the greatest problem before automotive engineers at present is the substitution of cheaper oils for gasoline as fuel in at least four distinct fields, namely, truck, tractor, motorboat and stationary farm engine. The passenger car field I omit because only in the others is saving in fuel cost necessary if we would not curtail their development.

If kerosene and the cheaper oils could be used entirely as fuel for the engines of all trucks, tractors, motorboats and stationary farm engines, millions of barrels of gasoline per year would be released for use in passenger cars, and not only that, but the quality of that gasoline could be much improved, owing to the fact that to a large extent the oil refiners would be relieved of their problem of how to dispose profitably of the large percentage of by-products they are obliged to make in order to get the gasoline.

Originally gasoline was a by-product of oil refining, for which there was comparatively small demand, kerosene for illuminating purposes being the all-important product. With the advent of the automobile and its offshoots the demand for gasoline increased with amazing rapidity. At the same time the demand for kerosene became less, owing to the use of electricity for illumination, even in rural districts. To-day the original conditions are reversed, gasoline being the all-important product and kerosene a by-product for which there is no correspondingly large demand.

Difficulties of Adapting Gasoline Engines to Kerosene

Unfortunately the kerosene yield from average crude oils by ordinary refining methods is about four times that of gasoline. It is obvious that the best interest of the oil refiners, as well as of the public, is served, not by the introduction of new and costly methods of refining, such as the cracking processes, but by adhering to the cheapest methods that will allow the maximum use of the products of oil refining. If, then, it be possible to use kerosene in large amounts in certain classes of engines, the necessity for cracking and blending processes will be lessened, and the quality of gasoline may be

improved in proportion to the amount of kerosene used.

The development of appliances for using kerosene in internal combustion engines has been very slow, and indeed there is not on the market today an entirely satisfactory device for burning kerosene in standard gasoline engines. It has been known for years that almost any gasoline engine will run after a fashion on kerosene, once the cylinders are hot. The knowledge of this fact led many engineers to work with the conventional devices that have proved satisfactory for carbureting gasoline, without giving much thought to the fact that kerosene and gasoline are widely different substances, having nothing in common except that they are derived from the same petroleum base. They are chemically different, their initial boiling points are widely divergent, that of good gasoline being 68 deg. fahr. and that of kerosene 338 deg. fahr., their boiling range is totally different, that of gasoline being 334 deg. fahr., while that of kerosene is 206 deg. fahr. Gasoline mixtures ignite spontaneously at about 680 deg. fahr., while like mixtures of kerosene self-ignite at about 575 deg. fahr. A permanent, fixed gas can be made by mixing good gasoline and air without the use of heat; with kerosene this is impossible.

In the methods of burning kerosene in conventional types of engines, one great essential stands out—*heat*. We must have heat to vaporize kerosene, and the intensity or amount of heat

must vary according to the loads put upon the engine. Now this is the greatest stumbling block with which we have to contend. After being started on gasoline and then run on kerosene when it has warmed up, the engine must supply the heat necessary to vaporize the fuel, so that it is out of the question to consider any further complication, such as the supplying of heat from an outside source. When the load on the engine is increased suddenly, more heat is needed to vaporize the additional fuel, and as this heat is generated by the combustion of fuel within the cylinders, there is a time lag before it is available.

At the February meeting of the Minneapolis Section of the Society the problem of burning kerosene in tractor

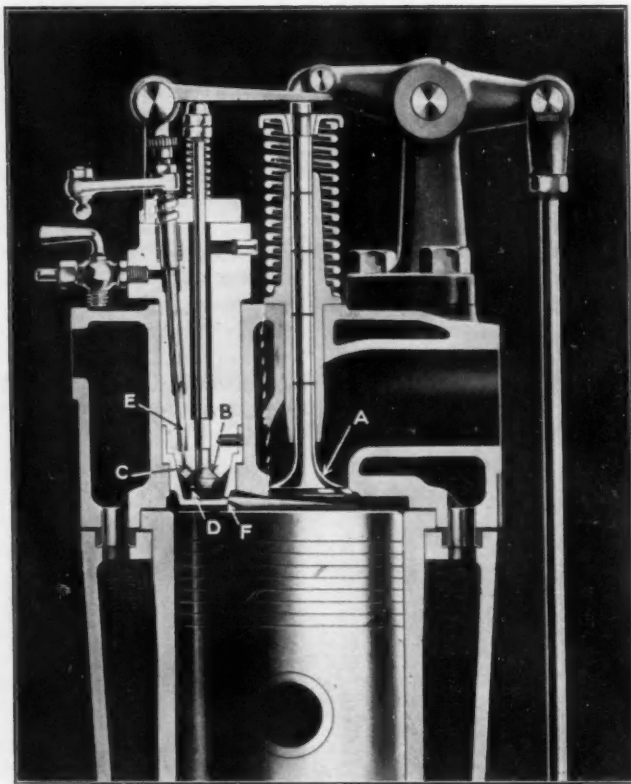


FIG. 1—CROSS-SECTION OF HVID ENGINE HEAD
Reference is made to the letters in tracing the cycle of operation

*Advisory Engineer, Sears, Roebuck & Company.

engines was discussed.* The opinion was then expressed that the ordinary hot-spot or heat-jacketed manifold did not solve the problem satisfactorily. The following points were brought out as a result of a questionnaire sent to 400 tractor owners in the state of Minnesota:

1 Only 40 per cent of these farmers recommended kerosene-burning tractors to their neighbors.

2 Many farmers complained that their tractors would not burn kerosene satisfactorily.

3 Dilution of the crankcase oil, owing to incomplete burning of kerosene and its seepage past the piston into the crankcase, was so excessive in some cases as to be dangerous.

In the light of these facts, it seems folly to try to burn kerosene in a gasoline engine, using practically the same device (the jet carbureter) as is used for gasoline with the application of heat and perhaps water injection.

I think absolutely satisfactory operation of a kerosene engine can be described somewhat as follows: First the

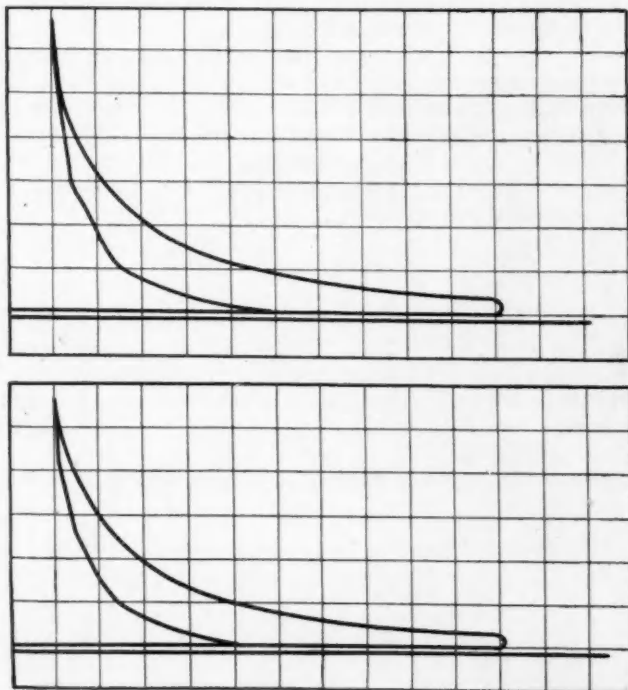


FIG. 2—INDICATOR CARDS OF 5% BY 9 IN. SINGLE-CYLINDER HVID ENGINE FROM WHICH FIGS. 3 TO 11 INCLUSIVE WERE OBTAINED

Indicator spring scale, one square represents 100 lb.

Upper Card		Lower Card	
80 lb. on scales		70 lb. on scales	
Area, sq. in.	0.55	Area, sq. in.	0.515
M. E. P.	87.3	M. E. P.	81.8
Max. pressure	670	Max. pressure	560

engine must start cold on kerosene without any delay or preheating; second, it must burn the fuel completely, without smoky exhaust; third, the fuel consumption must be as low as in the best gasoline engine; fourth, there must be no carbon deposits to foul spark-plugs or cause preignition; and fifth, it must follow closely what are accepted practices in design.

All the requirements thus cited can be met by an engine developed by R. M. Hvid, a naturalized Danish engineer. Not only will it permit the burning of kerosene and other heavier hydrocarbons, with high economy, but it does away entirely with all ignition and carbureting devices. Several companies are now making heavy duty engines of this type and it can also be made in light high-speed types.

*Fuels for Tractor Engines. By Prof. J. L. Mowry. THE JOURNAL, March, 1918.

OPERATION OF HVID ENGINE

Let us now examine the Hvid construction. In the first place, this engine operates on the ordinary stroke and is of conventional design (preferably of the valve-in-head type), weighing about 10 per cent more than a gaso-

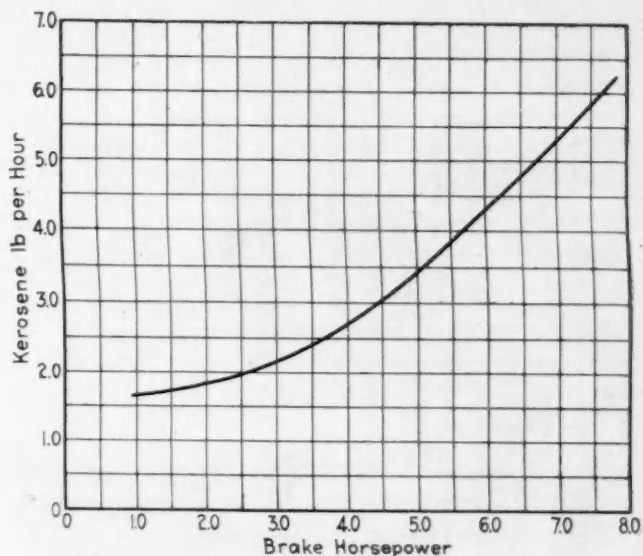


FIG. 3—FUEL CONSUMPTION CURVE. TEST NO. 1

Barometer, 28.91. First test indicated stratification of fuel and loss of 44 per cent through radiation with normal load, and 55 per cent with overload

line engine of the same rated horsepower. Ignition is secured by the heat of compression as in Diesel engines. For kerosene the compression is 390 lb. per sq. in.; for heavier oil it may be as high as 450 pounds. No compli-

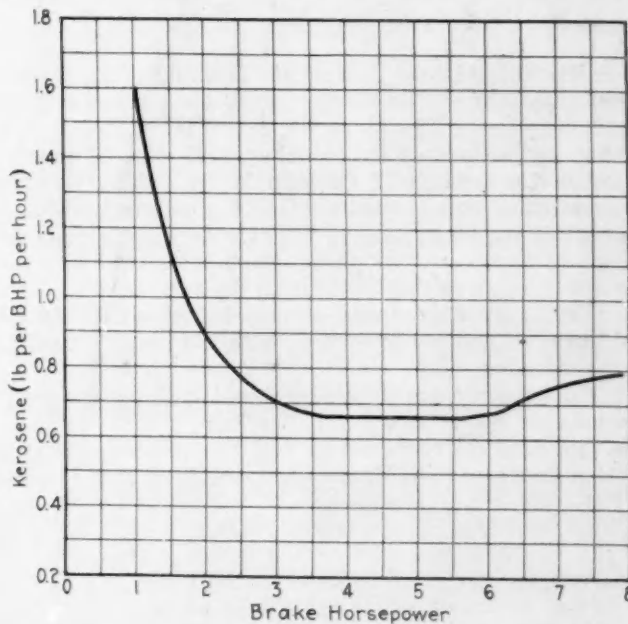


FIG. 4—FUEL CONSUMPTION PER B.H.P.-HR. TEST NO. 1

Barometer, 28.91. On the first test the lowest fuel consumption was 0.666 lb. per b. hp. per hr.

cated air compressor with its attendant piping is needed as in the Diesel engine, but fuel economy is on a par with Diesel practice. The cycle of operation (Fig. 1) is as follows:

1 On the intake stroke pure air only is admitted to the cylinder through a regular inlet valve A. While the air is

being drawn into the cylinder the fuel valve *B* is mechanically opened; some fuel flows out of hole *C* and drops to bottom of the steel cup *D*, the amount being controlled by needle valve *E*. At the end of the suction stroke fuel valve *B* closes and seals hole *C*.

2 The intake stroke is followed by the compression stroke, when the air previously admitted to the cylinder is compressed to about 380 lb. per sq. in., which renders

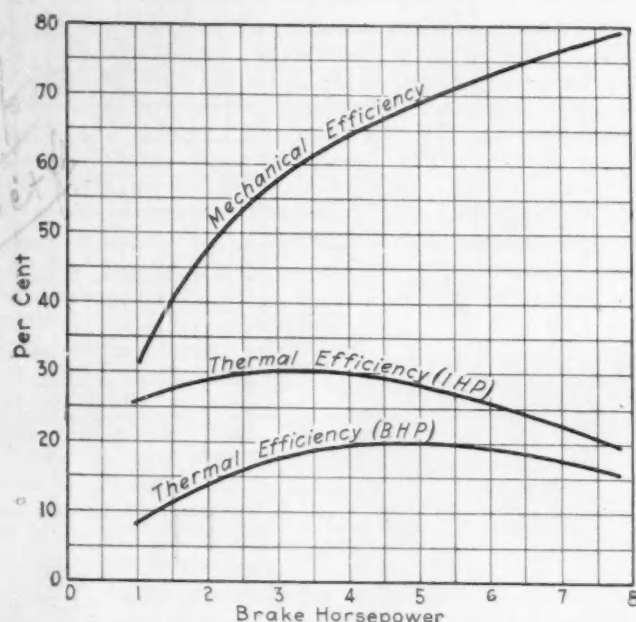


FIG. 5—EFFICIENCY CURVES. TEST NO. 1
Barometer, 28.91. During the first test the highest indicated thermal efficiency was 30 per cent at 3 hp, and highest brake thermal efficiency 20 at 5 hp.

it nearly incandescent. The air enters cup *D* through small holes *F* near its bottom, until the pressure in the cup is equalized. The air in the cup *D* is then red hot, as is that in the combustion chamber, and some of the fuel lying on the bottom of the cup is ignited by it. This combustion causes a sudden rise of pressure within the cup, far in excess of that in the cylinder, and a re-equalization of pressure takes place. By this time the air rushing out through the small holes in the cup carries with it the thoroughly heated and atomized fuel, which ignites and burns on coming in contact with the incandescent air in the cylinder.

3 The pressure due to this combustion forces the piston down on the power or working stroke.

4 The exhaust valve opens and the products of combustion are forced out by the upward stroke of the piston. The cycle is then repeated.

Here then is a design which actually does away with all the complications of ignition systems (the cause of about 85 per cent of all engine troubles) and of all intake manifolds and carbureters. Not only will an engine built on this design burn kerosene, but it will burn heavier and cheaper distillates within a wide range, without any adjustment. Its fuel economy, volume for volume, is nearly twice as great as that of the best gasoline engine. Its torque characteristics more nearly approach those of a steam engine (maximum torque at zero speed) than do those of any other internal combustion engine.

The disadvantage of the increase in weight of the Hvid engine over a gasoline engine of equal power, is negligible when compared with the many advantages. As a matter

of fact, many makers are now increasing the weight of their gasoline engines by strengthening the crankshafts and connecting-rods, because of the preignition pounding caused by poor gasoline.

The fuel situation is rapidly becoming such that the problem of burning kerosene in internal combustion engines will not be the only one; there will also be the problem of burning cheaper and heavier fuels in engines that are today burning gasoline. The fact that the Hvid design seems to solve these problems certainly entitles it to consideration.

A study of the results obtained from two dynamometer tests made recently at Armour Institute under the supervision of Prof. Daniel Roesch, of a 5¾ by 9 in. single cylinder Hvid type engine, will, no doubt, be of interest as showing its characteristics.

The engine was connected by two universal joints to an electric cradle-dynamometer. Engine speeds and fuel weights were obtained by means of electrically operated appliances. Arrangements were made for determining the jacket-water loss and the sensible heat of the exhaust gases. The developed and friction horsepower were determined by means of the dynamometer. Indicator cards were taken, but because of probable errors due to the high pressures involved and the comparatively high speed of the engine, these were used merely to study the valve settings and general events of the cycle and not to measure the indicated horsepower.

Preliminary Trials

The engine was operated under different loads and speeds and fuel supply, compression and cup design were varied. The final setting was made with 390 lb. per sq. in. compression.

Tests for Heat Balance

Test runs were conducted at a number of loads from

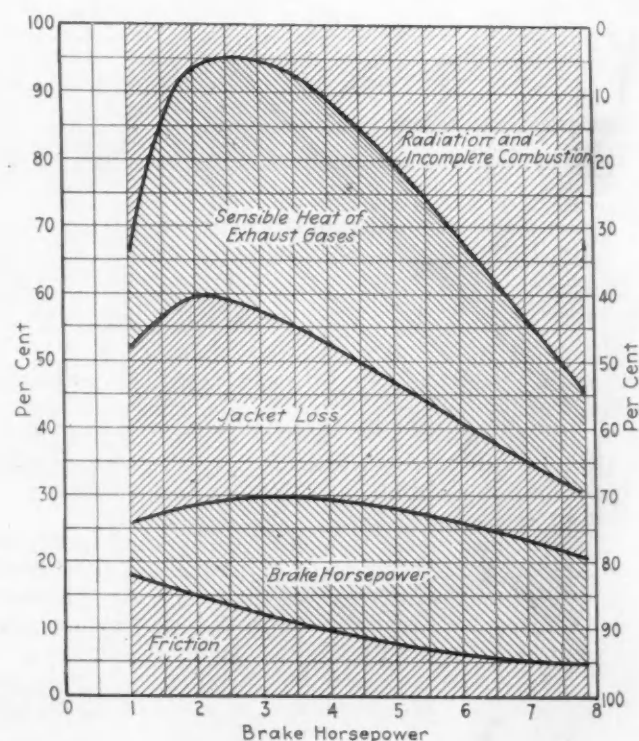


FIG. 6—HEAT BALANCE. TEST NO. 1
Barometer, 28.91. Chart of all readings, first dynamometer test

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maximum to about one-eighth of maximum, and readings taken to determine the following:

- 1 Friction horsepower (electric dynamometer method).
- 2 Brake horsepower (torque and speed).
- 3 Jacket-water loss.
- 4 Sensible heat in the exhaust.
- 5 Loss due to radiation and incomplete combustion.
- 6 Fuel consumption (pounds per hour).

Items 1 and 2 were determined directly from dynamometer readings. Items 3 and 4 were calculated from ob-

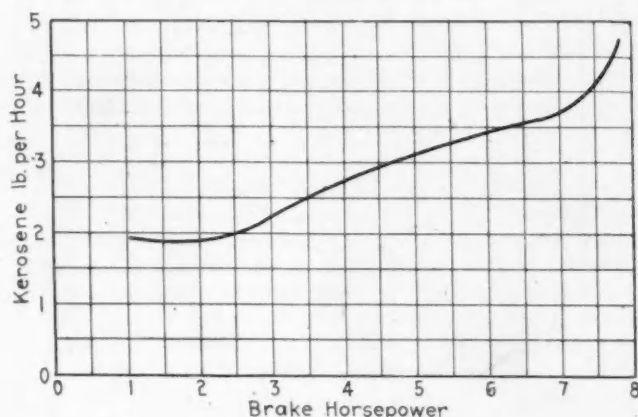


FIG. 7—FUEL CONSUMPTION CURVE. TEST No. 2

Barometer, 29.29. The second test indicated only 28 per cent of the fuel was lost in radiation and incomplete combustion, against 44 per cent on first test

served temperatures and weights. Item 5 was determined by the method of differences. Item 6 was obtained from direct measurements. The heat value of the fuel expressed in British thermal units per pound of kerosene was calculated from the formula

$$\text{B.t.u.} = 18,440 + 40 (\text{deg. Baumé-10}) \\ = 19,740 \text{ for the quality of fuel used in the tests.}$$

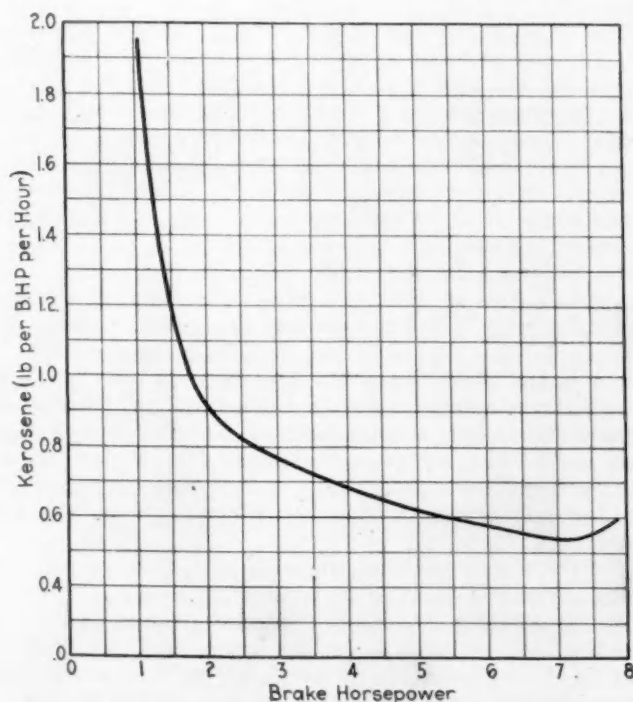


FIG. 8—FUEL CONSUMPTION PER B.H.P. PER HR. TEST No. 2

Barometer, 29.29. On the second test the fuel consumption dropped to 0.526 lb. per b.hp. per hr. with about the same load

The observed and calculated data are plotted in Figs. 2 to 11.

The first test runs indicated that stratification of the fuel was taking place in the combustion chamber; that when the engine was pulling its normal load 44 per cent of the fuel was lost through radiation and incomplete combustion; and that with 15 per cent overload as high as 55 per cent of the fuel was lost in the same manner. The engine, however, ran smoothly, developed all the power expected of it and showed wonderful torque characteristics.

After the first run the valve timing was changed slightly and the shape of the combustion space altered, so as to secure a better diffusion of fuel through the highly compressed air. The results then obtained on a second test were much better than the ones obtained on the first. One of the most important improvements was that of fuel consumption. At approximately normal load, only 28 per cent of the fuel was lost through radiation and incomplete combustion, as against 44 per cent on the first test.

On the first test the lowest fuel consumption was 0.666 lb. per b.hp.-hr., while on the second test, for practically

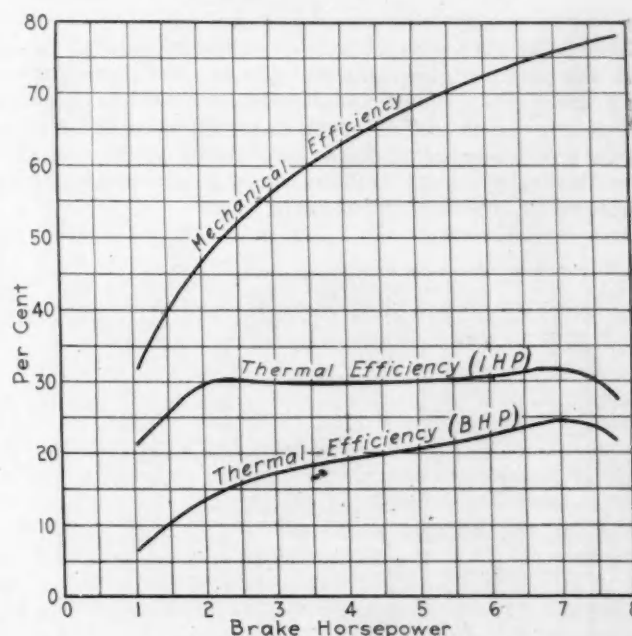


FIG. 9—EFFICIENCY CURVES. TEST No. 2

Barometer, 29.29. Thermal efficiency ran slightly over 32 per cent in second test and brake thermal efficiency was 25 per cent

the same load the consumption dropped to 0.526 lb. per b. hp.-hour. This is remarkable fuel economy for such a small unit.

During the first test, the highest indicated thermal efficiency was 30 per cent at 3 hp. and the highest brake thermal efficiency was 20 per cent at 5 b.hp. On the second test the indicated thermal efficiency was slightly over 32 per cent, and the brake thermal efficiency became 25 per cent, both at 7 hp., the rated power output of the engine. In other words on the second test the highest indicated and brake thermal efficiency was secured at the normal or rated power output of the engine, as should be the case.

Test for Torque

The test for torque was made by putting an arbitrary load on the engine, letting it run as fast as it would with this load, and then increasing the load and observing the

drop in speed, and also the brake hp. and the net mean effective pressure. At the start the load was 80 lb. on a 15 $\frac{3}{4}$ -in. radius, and the speed was 431 r.p.m., showing 8.62 b.hp. and 68 lb. per sq. in. net mean effective pressure. The load was increased by easy stages to 100 lb., at which point the speed had dropped to 272 r.p.m., but the brake horsepower at this speed was still 6.80 and the net mean effective pressure was 85 pounds. To put it in percentages, the speed dropped 37 per cent but the brake m.e.p. increased 25 per cent, so that the drop in b.hp. was only 20 per cent. This torque characteristic would be ideal for truck and tractor engines. The minimum test speed was not the minimum speed at which the engine would function well, but the limitation of low speed operation was reached because of excessive cyclical variation in speed which prevented accurate readings of torque.

I do not consider that the Hvid engine has by any means reached its ultimate stage of development, but the results already obtained, both with single and multi-cylinder types, have been so satisfactory that I believe all engineers should study this design carefully.

DISCUSSION

MR. GURNATT:—The Hvid engine has proved to us that the small engine can be built to burn kerosene or the heavier oils without complication and without a great increase in weight. The question is: Can the automobile engine be developed to burn the heavier fuels and still have the light weight and the speed range necessary in this type of engine? I believe it can.

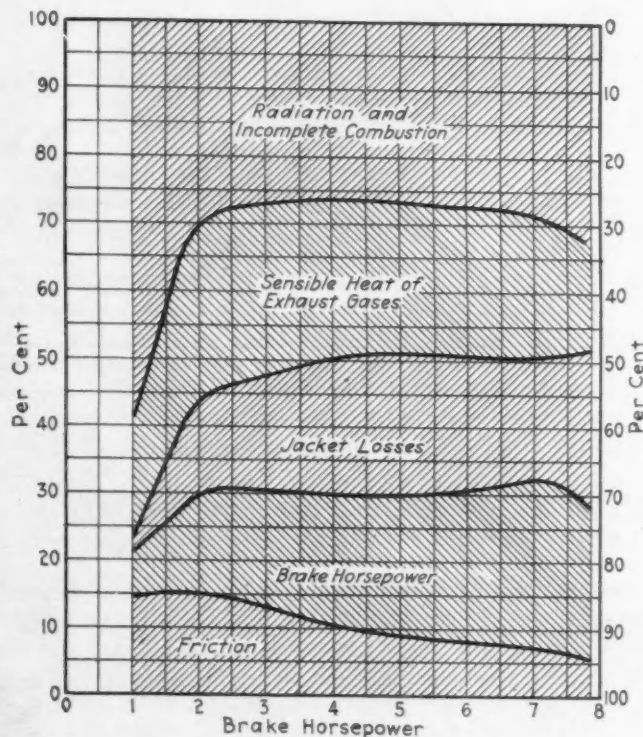


FIG. 10—HEAT BALANCE. TEST No. 2
Barometer, 29.29. Chart of all readings, second dynamometer test

Some years ago I had considerable experience with a rebuilt marine engine, single-cylinder type. The original head was removed and another one put on to permit operation on the constant pressure cycle. The fuel was admitted under pressure of about two pounds, and in starting a spark was required for ignition because we

could get only about 225 lb. of compression pressure. After the engine was warmed up the spark ignition was discontinued. The reciprocating parts of this engine weighed about 14 lb. and the flywheel weighed 150 lb. After the engine was hot it would burn a mixture of gasoline, alcohol, kerosene and lubricating oil from the

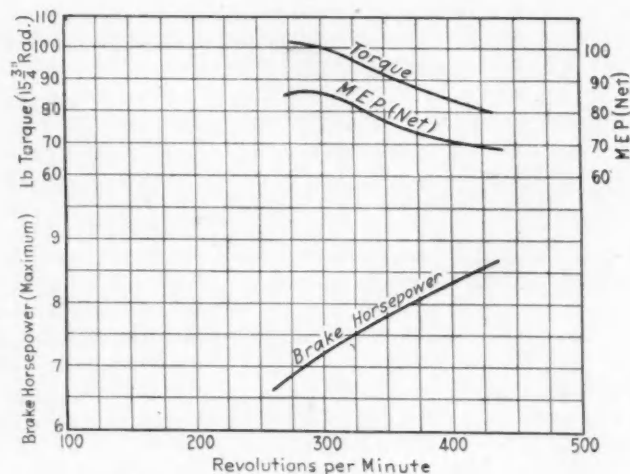


FIG. 11—TEST FOR TORQUE. TEST No. 1
Barometer, 28.91. In the torque test the r.p.m. decreased 37 per cent; brake m.e.p. increased 25 per cent; b.hp. decreased 20 per cent

crankcase—a very black oil. The performance was remarkable in that the speed increased from 100 to 2200 r.p.m. in less than three seconds.

Fuel Admission Mechanism

We had a few troubles with the engine. It would operate for a few days perhaps and then we would have to put in new fuel tubes and one or two new valves.

Has there been any trouble in the Hvid engine with the fuel holes clogging or fuel cups corroding and burning?

MR. BLAKELY:—Absolutely none. Pressure difference in cup and cylinder is great, and because of the high pressures the flow is so rapid through the holes, first in one direction and then in the opposite direction, that it would be impossible to clog the holes unless a leak developed between the packing of the cup and the fuel valve body; a deposit of carbon that would gradually plug the holes would then be built up in the cup. If the engine is operating perfectly and the cup joint is tight, such a thing as plugging the holes would be impossible. The cup never gets so hot that either corrosion or burning can take place. A little stream of fresh air from the outside is admitted into the cup, and the evaporation which takes place when the fuel hits the cup tends to keep it from getting too hot. In fact, the cups taken out immediately after a long run look blue, as though they had never been above a blue heat. It is possible to handle them immediately after a long run; to pick them up for an instant without burning or blistering the hands.

MR. GURNATT:—Have the size and location of the holes any direct bearing upon the fuel economy?

MR. BLAKELY:—Decidedly. We have found by experimenting that in order to get the lowest fuel consumption and the best power, the holes should be located slightly off the bottom of the cup, and so as to diffuse the oil or vapor streams as completely as possible through the mass of heated air. They should be as far apart as possible without permitting the streams to impinge upon any side

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walls. The fuel or vapor streams can strike the top of the piston with impunity, but the side walls of the ports or of the cylinders cannot be struck without losing power and fuel economy.

T. C. MENGES (M.S.A.E.):—The author states that when the suction valve *A* opens up, the valve *D*, (a fuel valve) opens at the same time, and as I understand it, the piston at that time is drawing in a charge of air. What prevents the liquid fuel from entering the cylinder at the same time?

MR. BLAKELY:—In the first place, unlike any other engine that we are familiar with, there is no choking effect in the intake passages. There is no manifold and no butterfly valve. The volumetric efficiency of the engine is always the same, the ports are wide open, and consequently if the valves are rightly proportioned the suction effect in the cylinder is very slight.

There is still another reason for keeping the holes above the bottom of the cup, as I explained before, when Mr. Gurnatt asked about their placing and the burning of the cup. A small stream of air passes down alongside the fuel admission valve; some of this undoubtedly, sweeps out through the holes when fuel is admitted, but the fuel dropping at the back and furthest away from the holes does not tend to go out. If any escapes there, the amount is very small. If the holes are not properly placed, or the size of the inlet valve is not proportioned rightly, fuel will be sucked in and there will be a harmful preignition.

Range of Suitable Fuel

EUGENE HIGGINS:—On what range in gravity of fuel would the conventional type of Hvid engine operate with given dimensions of the spraying apparatus? In other words, could we operate a truck or a tractor engine with a low grade of kerosene and then go to a fairly high grade of crude oil?

MR. BLAKELY:—Certainly. Operating with a range of gravity between 40 and 30 deg. no adjustment is required. Naturally the results are not as good with a 30 deg. crude oil when the engine is set to the best advantage for a 40-deg. kerosene, but the difference is extremely small.

We assume, of course, that an engine is built to run on a certain fuel. If it is to be kerosene, the engine will operate best on kerosene, but it is possible to operate within a wide range without making any adjustment, and the increase in fuel consumption will then be so slight as to be almost negligible.

Starting Is Easy

T. J. LITTLE:—Would there be any difficulty in the starting mechanism when applying this type of engine to an automobile? What would be the limits in the size of cylinder?

MR. BLAKELY:—In starting the engine, the compression pressure is zero so that getting up to speed would be really easier on an electric starter than with a type of gasoline engine in which the compression is not relieved.

So far as size is concerned, in a single-cylinder type, we find that the seven horsepower (the size I have described in the paper) is about the limit for hand-starting. In Chicago there is a four-cylinder tractor engine ($4\frac{3}{4}$ by $6\frac{1}{2}$ in.) which starts readily by hand.

Former Control Troubles Overcome

MR. MENGES:—It seems difficult to control the fuel supply. The governor on this engine controls its speed, and in the four-cylinder type a mechanism would be required to operate all four needle valves and at the same

time secure the same power from each of the four cylinders.

I have watched some of these stationary engines operating in the fields in the hands of farmers, and they seem hard to govern. After the engines are so adjusted that the fuel flow and mixture are correct, speed is then controlled fairly well until they begin to get hot. Then the governor does not seem to work properly. I have seen them race and run away; I have seen them get hot and have seen preignition occur, but have supposed it was the fault of the man at the engine.

Control of the speed and avoidance of preignition after the engine is under heavy load are two difficulties that are not yet solved. I am glad that they are being worked out, because there is a great field for this type of engine.

MR. BLAKELY:—Those difficulties certainly were true of some of the earlier engines. At first the attempt was made to control the fuel by a pointed needle valve; we discovered after a while that that was not the best way because the needle valve might tend to stick. The engine would then slow down unduly before the needle would release; there would be too much fuel rushing in and preignition would occur.

We have entirely eliminated that trouble now by means of a straight needle valve, with no point on it at all. The needle valve is sliced off on an angle, and slides up and down in its groove or hole. There is nothing to stick, and since we have adopted this type of needle, the control of these engines is closer than anything I have seen. Last week we made some tests on a heavy stationary engine about to be shipped to Washington. The governing was within five revolutions between no load and a 15 per cent overload applied suddenly or gradually.

In the four-cylinder engine a pointed needle valve was employed, but the means for operating it were different from those of the other engines that have been built. A screw or worm was used. Fig. 1 shows that the needle has no chance to stick as it would if it were operated straight up and down. An arm provided with a ball joint projects from the needle. There are similar arms on each cylinder and they are all hooked together. In order to adjust all four cylinders to operate alike we set the needle valves so that they close at a certain point, and then we have no trouble in governing at any load between shut-off and wide open intake.

C. S. RICKER (M.S.A.E.):—The engine referred to has a cylinder volume of 234 cubic inches. If the cylinder dimensions are increased we get a fairly sizable four-cylinder machine, suitable for a tractor or for a truck. How small a cylinder can we use and still measure the quantity of fuel injected? In other words, can we go down to 100 or 50 cubic inches?

Wide Limits of Size and Speed

MR. BLAKELY:—We are running very successfully and, so far as we can see, there is no minimum size limit. It is just a matter of measuring the fuel, and that is done by varying the "slice" on the straight needle.

I will admit that as long as we were using the pointed needle valve it was almost impossible to govern the small engines, but with the new needle valve, which in reality is not a needle any more, we have no difficulty at all. The engine is governed perfectly and runs steadily.

MR. RICKER:—What speed range can be obtained on these smaller engines?

MR. BLAKELY:—Unfortunately, the engine I have been describing is a low speed stationary engine, and the nearest approach I make in the paper to citing a speed range

is in the torque characteristic curves (Fig. 11) where the speed drops from 431 to 272 r.p.m., and the increase in the net mean effective pressure to take care of it is shown. I have run one of these little engines as high as 1300 or 1400 r.p.m. without a misfire. The four-cylinder tractor engine has been operated as high as 1200 and can be run as low as 180 r.p.m. either by throttling or overloading. This engine can be greatly overloaded without showing any distress. Naturally it is inefficient under such conditions; a great deal of fuel is wasted, but this happens with any other engine when it is overloaded. The only feature of the design that would limit the speed is weight of the reciprocating parts, and these would necessarily be a little heavier than in a gasoline engine of conventional type because of the somewhat higher pressure.

Timing of Ignition

S. M. WALKER:—A number of variables influence the time of ignition in all engines with which I am acquainted. These are the flash points of the fuel, atmospheric temperature, humidity of the atmosphere, the load imposed on the engine, the speed of the engine, and the altitude above sea level. How is the time of ignition controlled in this type of engine.

MR. BLAKELY:—At first glance, it would seem that this is not taken care of. It is true that the flash point of the fuel has a great deal to do with the control of ignition. All that is controlled, however, by the size of the holes in the cup. In replying to Mr. Higgins, I said that an engine will perform best with the fuel for which it was designed and adjusted, but will operate within a fairly wide range without any adjustment. The ignition is controlled to a large extent by the fact that when the engine is running at high speed the heat of compression is necessarily higher. Less heat is dissipated through the cylinder walls and by leakage of the gases past the piston rings, and ignition occurs earlier in the stroke when running fast than when running slowly. Apparently this takes care of itself nicely within a wide range.

It is true that altitude also has a great influence on the ignition, but so it has with every other engine. When this engine is overloaded it is not necessary to retard the spark manually because it retards its own spark down to a speed where a gasoline engine with a fixed electric spark would begin to pound and would eventually stop. This engine will keep on pulling until it gets to a speed where all the momentum of the flywheel is required to overcome the resistance due to the compression pressure.

Compression Pressures

MR. WALKER:—After an engine of this type has been in operation for say a year, what happens to the compression pressure? In the hands of the ordinary user is there not difficulty both in obtaining and maintaining such a high compression pressure?

MR. BLAKELY:—Many automobiles have been run for years, are still in good condition, and their engines are holding compression. When we talk about 390 and 400-lb. compression at the start we are startled. In the ordinary gasoline engine with say an initial compression pressure of 80 lb. On the explosion or combustion stroke the pressure rises to four times that, or 320 pounds. My experience has been that it is no harder to hold 400 pounds compression pressure than it is to hold 80 pounds, so that if we can hold one we can hold the other.

If a cylinder is distorted or out of round and piston rings do not fit, even 80-lb. pressure cannot be held. The

amount of wear on the piston rings is no greater in this type of engine than it is in a conventional type. It may be necessary to renew the rings after a year or two of service, as it is in a gasoline engine, but I do not realize any added difficulty over the gasoline engine.

MR. RICKER:—How much increase in the compression pressure is obtained on the explosion stroke?

MR. BLAKELY:—The indicator cards (Fig. 2) show that the maximum pressure is 670 lb. per sq. in. but that is unquestionably not a true card, owing to the high pressures in the indicator. We did not use those cards for anything except to study the valve timing and the events of the cycle. There is no doubt that much of the pressure indicated on the card is due to inertia. Ordinarily I would say that with 400 lb. compression pressure the explosion pressure would be perhaps 525 pounds.

MR. HIGGINS:—The obtaining of proper spark advance has been one of the great difficulties in gasoline engine development and correct operation. Can we accept as a fact the statement that the time of ignition is automatically advanced because of the increase in heat generated by the increase in speed?

MR. BLAKELY:—From a study of the indicator cards, which I have made by the hundreds, I would say that the statement is true. The results obtained are, after all, the only thing we can go by. They indicate that the moment of ignition is taken care of automatically and satisfactorily.

I failed to mention the fact that the cup gets a little hotter at high speed and for this reason also, the point of ignition is advanced. There is no chance for heat to be lost by radiation, and the cup is really hotter; this also helps to advance the time when ignition takes place. From all the results obtained by studying indicator cards, listening to the sounds of the engine and watching the exhaust, I believe that the time of ignition is very well taken care of.

MR. WALKER:—I think I understand what would take place under gradual load changes, but under a sudden change what would happen to the ignition?

MR. BLAKELY:—The four-cylinder tractor engine which I have mentioned, if suddenly throttled down from a heavy to no load, gives no indication of any necessity for spark adjustment.

E. W. ROBERTS (M.S.A.E.):—I have succeeded in running some engines nicely with kerosene and some poorly. I have seen engines run even on California oils, which according to my own analysis were 54½ per cent asphalt. What the Hvid engine will do on those I cannot predict, but I know that on some of the submarines off San Pedro, Cal., they have opened the crankcase after 24 hours of use and taken the residue out with a shovel.

Under expert operation an engine may work well on heavy oil but may not work well all that time. I have, however, actually seen, in the neighborhood of Los Angeles, a 125-hp. de la Vergne engine running at full load on 18-degree oil and with no smoke emitted from the exhaust pipe.

Unusual Oil Fuels

In some of the boats off Holland cod liver oil has been used in the Diesel engines. Twenty years ago engines in India were running on cocoanut oil and mustard seed oil.

I believe the Hvid engine is a solution of our kerosene problem, at any rate for fuels of gravity as low as 30 deg. Baumé.

MR. BLAKELY:—The Oregon, a ship recently launched on the Pacific Coast, was equipped with Hvid engines

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built by the Burnoil Engine Company, as auxiliaries for running the lighting and other equipment. There were two 50-hp. engines and a small 12½-hp. When the ship was about 150 miles off the Coast, the Diesel engines gave out and they ran back to port by using the Hvid engines for driving the air compressors, and using the compressed air to operate the Diesel engines. It was a slow proposition but the Hvid engines got the ship home.

MR. LITTLE:—What are the possibilities of increasing the volumetric efficiency?

MR. BLAKELY:—I really do not see how much one can expect to increase it. It is about as high now as it can be. It is limited only by the size of the inlet valve. It is true that if we could increase the volumetric efficiency we would get more power. If we could force the air in so as to fill the cylinders and so that the pressure would be slightly above atmospheric at the beginning of the compression stroke, we would get more power. That I think is a matter for future development.

MR. ROBERTS:—Oil engine operation depends on how much air can be forced in separately and on the amount that can be sprayed with the oil. When operating engines of the dry type in which the combustion chamber is a separate compartment it is necessary to make the clearance as small as possible. It is a principle well known to oil engine designers that as much of that air as possible must be forced in with the oil spray. There is no better illustration of the refinement to which this can be carried than the type FH de la Vergne engine.

MR. HIGGINS:—On account of the completely or partly congealed condition of the heavier oils, how much trouble would be encountered in starting in zero weather?

MR. BLAKELY:—We would have no trouble because the compression pressure would be arranged for this. Naturally, if we are going to use heavier hydrocarbons, the compression pressure must be higher. The lighter hydrocarbons come off first and give rise to the preliminary explosion that takes place in the cup. I have started a Hvid single-cylinder farm engine in zero weather on cylinder oil, the same oil I was using to lubricate the engine, without any difficulty, and surely there is not much of the lighter hydrocarbons left in lubricating oil.

FREDERICK PURDY (M.S.A.E.):—It seems to me, that the output of the engine is regulated by varying the quality and not the quantity of the fuel mixture.

Fuel Burned in Suspension

MR. BLAKELY:—There is no such thing as a mixture in this engine. The fuel is burned in suspension. The fuel, as it comes out in a spray or a gaseous state, burns the instant it strikes the red hot air in the combustion chamber. The quality of the mixture is not regulated. The engine output is regulated by the quantity of fuel. The fuel is burned in an excess of air, just as in a Diesel engine, consequently combustion starts when the fuel begins to emerge from the cup, and that is, of course, a matter of temperature.

MR. PURDY:—The Hvid engine under full compression all the time is equivalent to an ordinary throttled engine; the cylinder of the former always contains the full amount of air, and a limited amount of fuel is burned to produce the equivalent of throttling, as we understand it.

MR. BLAKELY:—There is no such thing in this type of engine as a "pop-back" or "flare back." If we had such a thing as a mixture in the cylinder when running with the fuel throttled it would be equivalent to a very lean mixture, and in a conventional type of engine that would

cause a pop-back. This is good proof that we are burning the fuel in suspension.

MR. PURDY:—This engine operates exactly on the Diesel cycle, so far as the burning of the fuel is concerned; with the Diesel type the speed and load cannot be varied as required for automobile service, consequently unless we depart in some way from the Diesel theory we must continue to use carbureters.

Flexibility of Hvid Engine

MR. BLAKELY:—The Diesel is a constant-speed engine which does not operate well at varying speeds. The Hvid engine is very flexible, not perhaps as flexible as a gasoline engine of refined design—we have not reached that point with the Hvid engine yet—but it differs in this respect from the Diesel engine. It can be operated at many more speeds than a conventional gasoline engine, and while its "pick-up" is not quite as rapid as that of a gasoline engine, this too is a matter for future experimentation and development.

MR. ROBERTS:—The mixture in the Hvid engine is burnt just as gas issuing from a jet is burned. So far as flexibility is concerned, if we go back some years to the early days of the automobile, when designers worked out all kinds of multiple-speed gears, in the belief that a gasoline engine must run at a constant speed at all times, we shall appreciate the progress evidenced in the Hvid engine. It bids fair to become as flexible as any automobile engine we have today.

Comparison with Brons Engine

MR. MENGES:—Just what is the difference between this and the Brons engine?

MR. BLAKELY:—The difference is this: The Brons has a series of holes completely around the cup and the pressure does not rise in the cup to eject the fuel as it does in the Hvid engine. Furthermore, in the Brons engine there is no method of controlling the fuel except by a globe valve, so that it is essentially a constant-speed engine. Brons' theory was that the fuel entering the cup would be ejected by a shaking due to the combustion of the fuel hanging by capillary attraction on the outside of the row of holes.

When Mr. Hvid applied for his patents in this country, only the Brons patents were cited against him so that he went to Holland and bought them.

Comparison with de la Vergne Engine

MR. AITKEN:—Mr. Blakely's description of the Hvid engine leads me to think that it will work equally well on light and heavy loads. Is that a characteristic of the de la Vergne type FH and DH engines?

MR. ROBERTS:—The characteristics are very different. It is difficult to run a semi-Diesel engine at much less than one-quarter load. In order to do it there must be some manipulation. For example, we have often had to choke the intake to make a semi-Diesel operate on increased load, and the semi-Diesel does not run satisfactorily on overload for any great length of time. The best work for a semi-Diesel engine is illustrated by the irrigation systems in California, where the pumping load is always equal to the rated load of the engine.

The economies of the Hvid engine are greater in the neighborhood of three-quarters to full load than they are on overload; economies at moderate load, theoretically at least, are greater than in any engine of the ordinary type because of the maximum compression at all loads.

MR. MACY:—Does much of the oil used in lubricating the Hvid engine break down under the pressure and temperature?

No Difficulty with Lubrication

MR. BLAKELY:—We have no difficulty with lubricants for the simple reason that the fuel is burned in suspension. There is no tendency on the part of the fuel to cut the lubricating oil. There probably is a slight destructive distillation of the film of lubricating oil that gets up into the upper ends of the cylinder, and this might be noticed if the engine were running at no load, particularly with a long exhaust pipe which would act as a condenser. There would be a slight amount of blue smoke coming out, no doubt the condensed vapor caused by the destructive distillation of the lubricating oil.

We went into the subject at Armour Institute when we were making our tests. We used a calorimeter for measuring the heat in the exhaust, and the water gave a good condensing action. When the engine was running at light load we could notice the slight blue haze. We disconnected the engine and calorimeter and the exhaust was as clear as the air; this convinced us that some destructive distillation was taking place, but we could not measure it.

We find too that very little lubricating oil is used. In fact, we are not using as much on our Hvid type engines as on our gasoline engines.

MR. AITKEN:—Does the analysis of the oil in the crankcase (I suppose that the system is of the pressure feed type and that the oil is carried in the crankcase) show that a heavier content of tar is washed down on account of the destructive distillation taking place?

MR. BLAKELY:—No. We never find any trace of

them. Whatever heavy tars or solid carbons are formed by the destructive distillation are thrown out in the exhaust. I had an interesting opportunity to examine two engines at our factory some time ago. One was a 7-hp. Hvid engine driving a lineshaft for operating machines. Sometimes four or five machines had to be started at once and sometimes for an hour there would be nothing to do. Outside was a throttling governor, so-called kerosene engine (gasoline and kerosene) which had been running under constant load for some six weeks. The 7-hp. Hvid inside had been running for seven months, and it happened that the cylinder heads of both were taken off the same day for different reasons. In the throttling governor engine cylinder were flakes of carbon as big as one's hand, red and discolored, and in the Hvid, around the engine, was just a slight, moist carbon deposit that could be wiped off with a piece of waste; the spot where the fuel streams strike the ends of the piston was absolutely clean.

MR. PURDY:—I have always thought that it costs something to compress air in a gas engine just as it does to compress it in an air compressor. If an engine is running with constant compression pressure, using the smallest quantity of fuel, does the energy required to compress the superfluous air equal the power developed by the fuel in the cup?

MR. BLAKELY: Fig. 6 shows that the mechanical efficiency of this engine is high, therefore it is evident that not much of the power is wasted to compress the air.

The engine from which these results plotted in Fig. 6 were obtained has heavy flywheels and other moving parts, and at no load the friction is so great that 2 hp. are required to run it. In spite of that, the mechanical efficiency is about 80 per cent, which is high.

Elements of Diesel Engine Design

By W. G. G. GODRON (Non-Member)*

MINNEAPOLIS SECTION PAPER

Illustrated With CHARTS

MORE and more prominence is given to the building of Diesel engines in this country, and several of the types built have, in many instances, proved to be very reliable and economical power units.

In 1915 about 400 or 500 plants were equipped with this type of internal combustion engine, a number which is increasing rapidly, even though the world war has kept back the production considerably.

After the war, however, the building of both stationary and marine types of Diesel engines will increase rapidly; already big shipbuilding companies in the United States are licensees of well-known European builders of Diesel engines, and, but for the entry of this country into the war, would be building these engines at present.

Although design, workmanship and material must be perfect, other factors of equally vital importance must be considered. One of these is the operator; the greater part of the trouble experienced is found to be directly due to the ignorance and carelessness of the operator and

the use of lubricating oil not suitable for the severe work to which the engine is subjected.

In order to be allowed to operate a steam engine, an engineer's license is required; this is given only when an examination is successfully passed, assuring intimate knowledge on the part of the engineer of the engine under his charge. For the operation of a Diesel engine no license is required by law, although it is more intricate and difficult to keep in good working order. A mere trifle that would not affect the operation of a steam engine will stop a Diesel engine.

The fact that no license is required to operate a Diesel engine will be the source of many troubles for which the engine will be blamed unjustly.

Diesel engines are designed both as four-stroke and two-stroke cycle engines, and work on what is known as the constant-pressure cycle, that is, the combustion takes place at constant pressure with previous compression.

Diesel engines are developing rapidly, and although at this time the four-stroke cycle type is used principally for stationary work up to 1000 hp. and the two-stroke

*Chief, Technical Department, South American Branch of Vacuum Oil Company, Buenos Ayres.

cycle type for larger engines, it is the belief of many builders that the double-acting, two-stroke cycle may, in later years, be used as a standard type for all powers.

In order to get a clear idea of the principle of operation, we will first go over the diagram (Fig. 1), representing the work done in a four-stroke cycle gasoline engine. Here is no slow combustion as in the Diesel engine, but an explosion at constant volume.

The first or suction stroke, is represented by the horizontal line (Fig. 1.), the piston moving away from the head of the cylinder and drawing in the explosive mixture of gas and air through the inlet valve, the pressure remaining nearly at atmospheric.



FIG. 1 — FOUR-STROKE CYCLE EXPLOSION ENGINE DIAGRAM

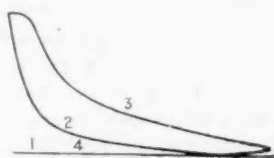


FIG. 2 — FOUR-STROKE CYCLE DIESEL COMBUSTION ENGINE DIAGRAM



FIG. 3 — TWO-STROKE CYCLE DIESEL COMBUSTION ENGINE DIAGRAM



FIG. 4—SAME ENGINE AS FIG. 2 WITH FUEL VALVE OPENING TOO EARLY

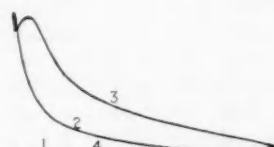


FIG. 5—SAME ENGINE AS FIG. 2 WITH FUEL VALVE OPENING TOO LATE

The second or compression stroke is represented by line 2, the piston returning toward the head of the cylinder, and, all valves now being closed, compressing the combustible mixture in the cylinder to say 60 lb. per square inch.

The third or power stroke is represented by line 3; first ignition of the combustible mixture by electric spark which gives an explosion, bringing the pressure above the piston up to say 300 lb. per sq. in., is shown by the vertical line, followed by the piston moving away from the head of the cylinder through the expansive force of the gas.

During the fourth or exhaust stroke, represented by line 4, covering line 1, the piston moves toward the head of the cylinder, driving the exhaust gases out through the now opened exhaust valve at nearly atmospheric pressure.

As will be seen from the above, only two valves are necessary, an inlet and exhaust valve, and an electric spark-plug to ignite the combustible mixture at the right time.

FOUR-STROKE CYCLE DIESEL ENGINES

Fig. 2 represents the work done in the cylinder of a four-stroke cycle Diesel engine. Here also line 1 repre-

sents the first or suction stroke. The air inlet valve is open while the piston moves away from the cylinder head, drawing pure air into the cylinder under atmospheric pressure instead of a combustible mixture of gas and air, as is the case in a gasoline engine.

Line 2 represents the second or compression stroke. The air inlet valve is closed and the piston, moving toward the head of the cylinder, compresses this air to a pressure of about 500 lb. per sq. in., reaching thereby a temperature of about 1000 deg. fahrenheit.

At about 8 deg. before the piston reaches the end of the second or compression stroke, the fuel valve opens and fuel oil is injected through an atomizer into the cylinder by means of compressed air of sufficient pressure to break the oil into the smallest particles possible before it enters the combustion chamber, where the hot compressed air of 500 lb. per sq. in. pressure and 1000 deg. fahr. temperature ignites and supplies the oxygen for the complete combustion of this oil.

The fuel valve remains open for 10 to 12 per cent of the stroke, during which time the combustion takes place. During the rest of the stroke, all valves being closed, expansion takes place, as represented by line 3 on the diagram, the piston moving away from the head of the cylinder.

Line 4 represents the exhaust stroke, the piston moving toward the cylinder head and driving the exhaust gases out of the cylinder through the now open exhaust valve.

When the piston reaches the end of its stroke, the exhaust valve closes again, the air inlet valve opens and the cycle is repeated. As will readily be seen, four strokes of the piston or two revolutions of the crankshaft are necessary in order to get one power stroke, hence the name four-stroke cycle engine.

TWO-STROKE CYCLE DIESEL ENGINES

Fig. 3 represents the work done in the cylinder of a two-stroke cycle Diesel engine.

Line 1 represents the compression stroke, during which the pure air is compressed, the same as in the four-stroke cycle, to about 500 lb. per sq. in. and about 1000 deg. fahrenheit.

During the second and last stroke the work is the same as is done by three strokes in the case of the four-stroke cycle Diesel engine represented in Fig. 2 by lines 3, 4 and 1, that is, combustion, expansion, exhaust and air inlet.

Combustion and expansion are the same as in the four-stroke cycle, except that the expansion is cut off earlier, and during the rest of the stroke exhaust and air admission take place.

At point A on line 2 the piston uncovers exhaust ports in the cylinder wall, the pressure in the cylinder dropping rapidly to atmospheric pressure. Then the scavenging air valve in the cylinder head is opened and air under a pressure of about 4 to 6 lb. rushes into the cylinder, blowing the remaining exhaust gases out through the exhaust ports and at the same time filling the cylinder with pure air. The cycle is then repeated; the piston moving toward the cylinder head covers the exhaust ports, the scavenging valve closes and pure air is compressed.

In some makes of Diesel engines the scavenging ports in the cylinder wall opposite the exhaust ports take the place of the scavenging air valve in the head of the cylinder and the piston head is then shaped in such a manner that the scavenging air rushes first toward the cylinder head and then turns toward the piston, driving the

exhaust gases out through the exhaust ports. In this case, only one valve, the fuel valve, is in the cylinder head.

Although the lines 1 and 4, Fig. 2, show atmospheric pressure, in reality 4 is slightly above, while 1 is slightly below atmospheric pressure, owing to the driving out and drawing in respectively of exhaust gases and air. This little area between those two lines, representing negative work done, is not perceptible on the diagram. As during the injection and combustion of the fuel the piston moves away from the head of the cylinder, the in-

is usually pumped up to pressure again immediately after starting. The air pressure used for starting varies with different makes of Diesel engines and is from 300 to 1000 lb. per sq. in. For starting large engines about 350 lb. air pressure is usually needed.

The pressure of the blast or injection air usually varies with the load on the engine as well as with the various types of engines and is from 650 lb. with light load to 1100 lb. with full load. Some manufacturers use practically the same air pressure for all loads.

The air compressor is usually driven from the engine crankshaft, and, except for small engines, compresses the air to the required pressure of say 1000 lb. per sq. in. in three stages.

Fig. 6 shows a diagrammatical sketch of a three-stage air compressor as used for Diesel engines and driven direct from the crankshaft. The air under atmospheric pressure is drawn in the first stage or low-pressure cylinder and is then compressed to about 45 lb. This air is hot and passes through an intercooler. From there the now cold air is drawn into the second stage or intermediate pressure cylinder and compressed to about 200 lb.; it again rises to high temperature and is again cooled in an intercooler, and is then finally drawn into the third stage or high-pressure cylinder, where it is compressed to the required pressure of about 1000 lb. per sq. in. The air, under this high pressure, is again cooled and forced into the blast or injection air bottle. Another pipe leading from this injection air bottle connects with the fuel valve in the power cylinder cover and furnishes the air necessary for the atomization of the fuel.

The air compressor piston is so designed that the temperatures of the compressed air, after each stage of compression, are about the same. As the atmosphere always contains more or less moisture which condenses in the coolers after the various stages of compression, sufficient drain cocks are placed at suitable points in the air line to prevent the accumulation of water in the air compressor and air bottles.

As the temperature of the compressed air in each stage of compression depends on the design, cooling and atmospheric conditions, it can be as high as 500 deg. Fahr., but in case of leaking valves, where part of the hot air flows back into the cylinder during the suction stroke, the temperature may rise considerably above this point.

Because of these high temperatures, it is of the utmost importance to reduce the amount of lubricating oil for the air pistons to as low a point as possible in order to prevent excess oil from passing the air valves and carbonizing there. Many of the troubles with air compressors can be traced to the use of the wrong kind of lubricating oil, and to overfeeding even when the correct high-grade oil is in service. Explosions in air compressor cylinders and cooling coils can be traced to excess feeding of lubricating oil, the oil "cracking" and forming an explosive mixture.

Good results are obtained with high-grade mineral oils applied very sparingly in the right places. Other troubles, however, sometimes experienced with air compressors of Diesel engine units, are attributed to the lubricating oil used, although this may be absolutely blameless; these are:

- (a) Dirty surroundings, permitting dust to enter the low-pressure cylinder during the suction stroke.
- (b) Chemicals in the air entering the compressor, which attack the material, especially the copper cooling coils.
- (c) Failure to drain the air line at sufficiently fre-

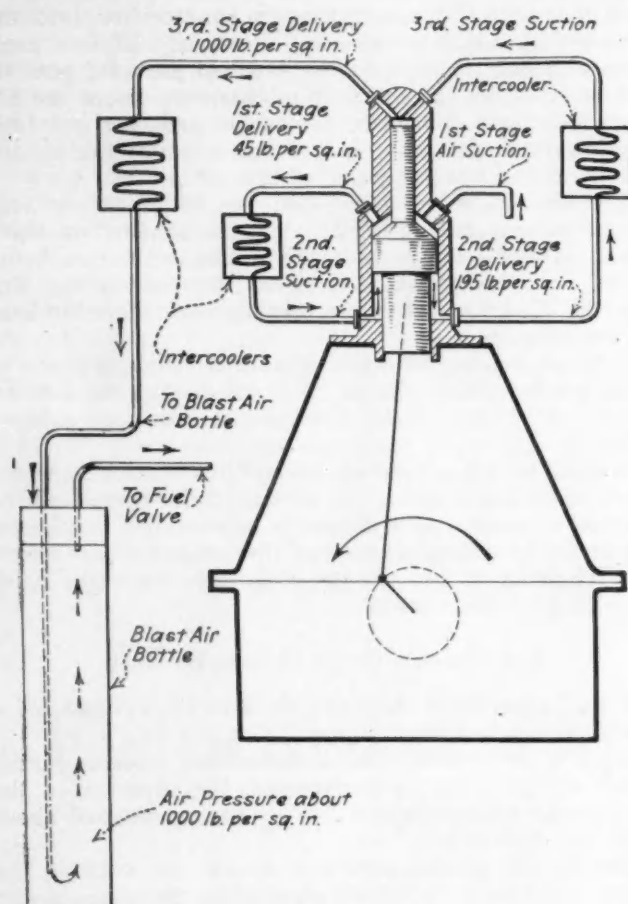


FIG. 6—PRINCIPLE OF THREE-STAGE AIR COMPRESSOR FOR A DIESEL ENGINE

crease in pressure, owing to the combustion, is nearly compensated by the increase in volume, so that the pressure remains practically the same during this period.

As will be seen from the foregoing, three valves are necessary when the four-stroke cycle Diesel engine is in operation; that is, the fuel valve, exhaust valve and air inlet valve respectively, and these are all placed in the cylinder head or cover. There is also a fourth valve, the starting air valve, used only when starting the engine. Compressed air from a reservoir is used for this purpose. This valve closes automatically when the engine acquires sufficient speed to ignite the fuel charge by means of the high temperature of the compressed air and continues to run on fuel.

COMPRESSED AIR FOR TWO PURPOSES

Compressed air is needed for two purposes, for starting the engine and for supplying the so-called blast or injection air. The starting air is used only when starting the engine and is taken from a special reservoir, which

quent intervals, especially when the humidity of the air is high.

(d) Leaking air valves resulting in too high a temperature of the compressed air.

FUEL VALVE

In order to have complete combustion in a Diesel engine it is necessary that the injected fuel should be burnt while it is in suspension in the air, and for that reason the piston tops are often concave so as to give the fuel a longer distance to travel and more time for burning. This prevents the oil from depositing on the piston head, where it would be cooled to such an extent by contact with the metal that incomplete combustion would result; carbonized oil would then be deposited on the piston head.

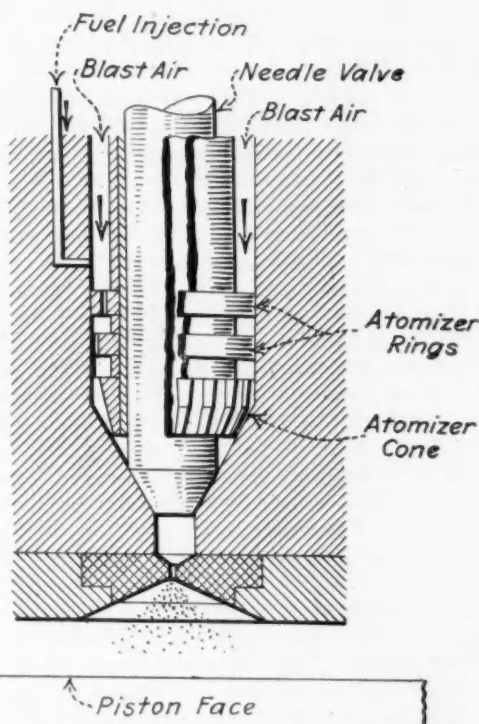


FIG. 7—CLOSED NOZZLE TYPE FUEL VALVE

There are two distinct types of fuel valves, the closed nozzle and the open nozzle. The latter has been used thus far for horizontal engines only. The various makes of Diesel engines differ greatly, especially in the manner of atomizing the fuel.

Fig. 7 shows a closed nozzle type of fuel valve known as the Sabathe valve. Before the needle valve opens, the fuel pump has pumped the necessary quantity of oil through the narrow passage on the top of the atomizer rings, which fit closely in the fuel valve box. Two rings are shown here; very small holes are drilled near the inner edge in the top rings and in the bottom ring near the outer edge, the number of these rings depending on the kind of fuel used, and increasing when a lighter fuel is used.

The circular space around the needle valve and between the rings is always under a pressure from the injection or blast air of about 1000 lb. per sq. in., so that the fuel has to be pumped into this space against this high pressure. When the needle valve opens the air forces the oil at great velocity through the small holes in the rings and through the atomizer cone or sprayer in the form of a spray into the combustion chamber, where it comes in contact with the highly compressed (600 lb. per sq. in.) and hot (1000 deg. Fahr.) air and complete combustion takes place.

FUEL PUMP

The amount of fuel injected is regulated by the governor, usually acting on the suction fuel valve, holding this valve open for a longer time when working on light load and bringing it to its seat sooner when working under heavier load.

As the amount of fuel oil injected is very small, especially when working on light load, it is of great importance to keep valves and stuffing boxes of the fuel pump perfectly tight in order to inject the same amount of fuel into each cylinder.

Some manufacturers design fuel pumps with one plunger and use distributors to give each cylinder its

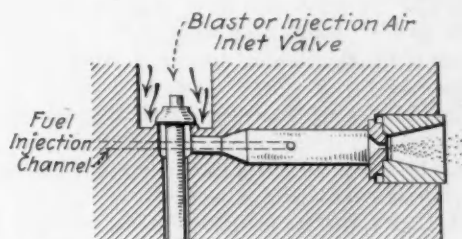


FIG. 8—OPEN NOZZLE TYPE FUEL VALVE

correct quantity of fuel, while others use a separate plunger and valves for each cylinder.

Before starting a Diesel engine, it is necessary to make sure that there is no air in the fuel line up to the fuel valve body and a hand pump is usually provided to pump the oil as far as the fuel valve where it is by-passed into an oil trough, all air being driven out in this manner. When the engine is started on compressed air the first stroke of the fuel pump will force the fuel oil through the atomizer rings.

The open nozzle type of fuel valve (Lietzenmeyer), as shown in Fig. 8, is used in many types of horizontal Diesel engines. In this design the place where the fuel is pumped is always in connection with the cylinder between the injection or blast air valve and the combustion chamber, and thus not always subjected to the heavy blast air pressure as is the case in the closed nozzle valve.

In this case the fuel pump does not pump the fuel against high blast air pressure but at the time when there is no pressure in the cylinder. When the blast air valve opens, the air strikes over the surface of the oil on its way to the cylinder, by this action atomizing the oil and blowing it in the form of spray into the cylinder.



SPECIFICATIONS OF LIGHT U. S. ARMY TRUCK

THE Motor Transport Service of the Quartermaster Corps of the U. S. Army has recently announced the specifications of the new standard light cargo-truck chassis, with capacity of $\frac{3}{4}$ to 1 ton. This chassis will take the place of the Class AA military truck chassis which was designed last summer by a group of engineers. The specifications are listed below:

DETAILED SPECIFICATIONS OF CHASSIS

General Dimensions

Wheelbase	in. 132
Length overall (including bumpers)	in. 188 $\frac{3}{16}$
Length of frame back to toe board, standard	in. 109
Overhang from center of rear axle to end of frame	in. 32
Maximum outside body width between rear wheels	in. 46 $\frac{3}{4}$
Tread of front and rear wheels	in. 56
Turning radius	ft. 23
Road clearance under front axle	in. 10 $\frac{3}{4}$
Road clearance under rear axle	in. 8 $\frac{3}{4}$
Total width of hubs	in. 68
Load height of frame, front	in. 26 $\frac{1}{4}$
Load height of frame, rear	in. 27
Rear spring, length	in. 54
Rear spring, width	in. 2 $\frac{1}{2}$
Front spring, length	in. 38
Front spring, width	in. 2 $\frac{1}{4}$
Tires, front and rear	in. 35 x 5
Frame width at rear	in. 34
Weight of chassis (approximate)	lb. 2800

Front Axle

- Drop Forgings I section, 2 $\frac{5}{8}$ in. high by 1 $\frac{3}{4}$ in. wide, heat-treated
- Steering Knuckles and knuckle arms drop-forged and heat-treated
- Knuckle Bolts and tie-rod bolts hardened and ground, and turning in hardened bushings
- Knuckle Tie Rod adjustable, provided with locking device on threaded end
- Ball Pin is steel, heat-treated, hardened and ground to true sphere, 1 $\frac{1}{8}$ in. diameter. Ball pin has taper shank, fitting taper hole in knuckle arm

Front Wheels

- Artillery type, S. A. E. Standard, hickory spokes. White oak, hickory, rock elm or ash felloes. Twelve spokes per wheel
- Spokes, oval section, slightly larger at outer ends
- Bearings, cup-and-cone type, ball bearings
 - Outer, 10 balls of $\frac{5}{8}$ in. diameter
 - Inner, 12 balls of $\frac{3}{4}$ in. diameter
- Hubs of malleable iron. Flanges of pressed steel, secured by eight bolts
- Hub Caps nickel plated

Rear Wheels

- Same size and design as front wheels, except as to hub bore and drilling
- Bearings, double-row ball bearings, both radial and thrust. One bearing 5 $\frac{1}{8}$ in. diameter, 2 $\frac{3}{8}$ in. bore, and 2 $\frac{1}{8}$ in. wide, containing two rows of balls, each row eleven (11) balls, $\frac{3}{4}$ in. diameter
- Hubs, malleable iron, secured to wheel by twelve (12) bolts, six (6) of which also hold driving flange. Felt washers to retain grease in the bearings. Oil shedders on inner edge of hub
- Brake Drums, pressed steel, 3/16 in. thick, secured to spokes by same twelve (12) bolts that hold hubs in place. Drum 14 in. diameter by 2 in. face
- Shafts of nickel steel, heat-treated, 1 $\frac{5}{8}$ in. diameter (inner) and splined, forming six (6) keys fitting differential hub, the outer end of which shaft is tapered

and keyed to the rear wheel flange, which in turn is secured by the six (6) bolts previously noted

Tires and Rims

- Front and rear, 35 by 5, pneumatic, straight side Q. D., carried on 34 by 4 $\frac{1}{2}$ in. Baker demountable rims.
- All dimensions S. A. E. Standard
- Goodyear, U. S., or equal fabric cord

Springs

(Half-elliptic, front and rear)

Dimensions—Front springs		Dimensions—Rear springs	
Length	in. 38	Length	in. 54
Width	in. 2 $\frac{1}{4}$	Width	in. 2 $\frac{1}{2}$
Depth at center	in. 2 $\frac{1}{8}$	Depth at center	in. 3.9
Number of leaves	8	Number of leaves	15

Rebound leaf, used on rear springs only, is placed above the main leaf, all leaves high-carbon steel except top main leaves at rear springs, which are alloy steel, heat-treated

Spring Bolts, hardened and ground, recessed for lubrication and provided with compression grease cups on each bolt and shackles. Spring bolts tapped into one jaw of shackle and provided with lock nut

Spring Clips, round, heat-treated, alloy steel, front $\frac{5}{8}$ in., rear $\frac{3}{4}$ in. diameter, secured by nuts and lock washers. A spring clip plate is used on top of spring, and spring rests on block, which in turn rests on spring seat on the axle. These blocks and seats are machined

Front Spring Shackle Bolts turn in bronze bushings in frame bracket. Rear shackles are bronze bushed, and swing on steel bearings

Steering Gear

Screw and nut type, semi-reversible and adjustable. Case is of malleable iron. All working parts ground to accurate finish. The screw is of hardened steel, and the split nuts of semi-steel, with hardened steel shoes attached to the end. Steering arm and trunnion shaft are drop-forged steel, heat-treated. Gear is anchored in bracket riveted to the frame and stiffened by a plate and collar on the floor board. Steering arm is secured to splined and tapered end of shaft by nut and lock washer

Shaft arm revolves in renewable bronze bushings; ball is 1 $\frac{1}{8}$ in. diameter, heat-treated. Steering wheel is 18 in. diameter, with laminated wood rim. Spider is malleable iron, with flat arms. Spark and throttle levers are mounted on top of the steering wheel. These levers are marked so as to distinguish between them, and engage quadrant, on which proper markings show the advance of the spark and opening of the throttle. All control tubes are bushed, spark and throttle control gears are of bronze or malleable iron. Steering rod is fitted with spring sockets, which are lubricated by compression grease cups

Brakes

There are two (2) sets of brakes, both on the rear wheels Service Brakes, external contracting type, operated by foot pedal. Brake lining of copper asbestos fabric Diameter of brake band, 14 in. Width of brake band, 2 in. Each band is provided with wing nut adjustment

SPECIFICATIONS OF LIGHT U. S. ARMY TRUCK

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Emergency Brakes, internal expanding type, operated by hand lever, which is fitted with ratchet lock. Brake lining is copper asbestos

Diameter of brake shoe, $13\frac{1}{2}$ in.

Width of brake shoe, $1\frac{3}{4}$ in.

Brake Adjustments are made by self-locking wing nuts on the brake rods. Equalizers are not used

Transmission

Sliding gear selective type. Three forward speeds and reverse, direct drive on third speed. Unit mounting with clutch and engine

Shafts are of vanadium alloy steel, heat-treated. Main shaft, $1\frac{7}{8}$ in. diameter, carrying six (6) splines for sliding gears. Countershaft, $1\frac{1}{4}$ in. diameter

Gears are of vanadium alloy steel, heat-treated. Sliding gears are splined to main shaft. All countershaft gears are secured to the shaft by four (4) round keys. All gears are 5-7 pitch and $\frac{7}{8}$ in. face

Ball Bearings. Pilot bearing on main shaft, double row type, lower row of bearings single row type

Case is of cast iron. Stuffing box on main shaft.

Breather in cover. Oil filler elbow on side of case

Control Levers mounted directly on transmission. All levers, forks, etc., are drop-forged steel. Transmission lever is topped with metal ball threaded and locked to lever

Gear Ratios

FIRST—		THIRD—	
Transmission	3.65	Transmission	1.0
Bevel drive	6.0	Bevel drive	6.0
Total gear reduction	21.9	Total gear reduction	6.0
SECOND—		REVERSE—	
Transmission	1.58	Transmission	4.43
Bevel drive	6.0	Bevel drive	6.0
Total gear reduction	9.48	Total gear reduction	26.58

Power Take-Off

Transmission is supplied with inspection plate upon removal of which Kellogg twin-cylinder tire pump can be attached on the right-hand side

Radius Rods

Thrust from rear axle frame is taken by radius rods of steel tubing. They are provided with ball and socket joints on each end, with springs for absorbing shocks. Ball pins and sockets are hardened and provided with compression grease cups

Engine

Four-cylinder, four-stroke cycle, vertical, water-cooled
Bore, $3\frac{3}{4}$ in. Stroke, 5 in. S. A. E. (A. L. A. M.), hp., 22.5.

Piston displacement, 220 cu. in.

Crankcase is aluminum casting. Oil pan steel stamping.

All bearings are carried on the crankcase

Cylinders L Type, cast in block. Water jacket heads cast separately

Pistons are $3\frac{3}{4}$ in. long, carrying three (3) rings and necessary oil grooves. All pistons and connecting rods to be carefully weighed and balanced

Piston Pins of steel tubing hardened and ground to size. Bearings are bronze bushings pressed into connecting rods

Connecting-rods, I-beam section, 0.35 carbon steel, drop-forged and heat-treated. Connecting-rod caps held in place by two (2) nickel-steel bolts with nuts properly locked. Connecting-rod bearings, 2 $\frac{3}{16}$ in. long, $1\frac{7}{8}$ in. diameter

Crankshaft, drop-forged and heat-treated to tensile strength of 90,000 lb. per sq. in. Crankshaft is carried on three (3) bearings

Front $2\frac{7}{8}$ in. long by 2 $\frac{3}{16}$ in. diameter

Center, $3\frac{1}{2}$ in. long by 2 $\frac{7}{32}$ in. diameter

Rear, 3 in. long by $2\frac{1}{4}$ in. diameter

Thrust flanges are provided on both ends of the center bearing

Flywheel, 15 in. diameter; $3\frac{5}{8}$ in. face

Bearings of crankshaft and connecting-rods of bronze, lined with highest grade hard babbitt. Bearings are held in place by brass retaining screws. Adjustment is secured by sheet-steel shims

Valves, $1\frac{7}{8}$ in. diameter, stems and adjustable tappets are enclosed. Valves have nickel-steel heads, electrically welded to carbon-steel stems. Valve stem ends are hardened

Push rods, mushroom type, hardened and ground all over. Adjustable tappet screws in tops of push rods. Push rods operated in removable guides and held in place by forged clamps

Camshaft, low carbon steel, drop-forged, heat-treated and ground. Camshaft provided with three bearings of white bronze of the following sizes:

Front, $1\frac{1}{2}$ in. long by 2 $\frac{19}{32}$ in. diameter

Center, $1\frac{1}{8}$ in. long by 2 $\frac{37}{64}$ in. diameter

Rear, $1\frac{3}{4}$ in. long by $1\frac{1}{2}$ in. diameter

Lubrication, combination of force feed and splash

Manifolds, exhaust and intake manifolds cast separately

Clutch

Multiple disk, dry plate type, five (5) driving plates lined on both sides with "raybestos" and six (6) driven plates not lined. The drum of the flywheel carries 106 driving teeth. Clutch shaft is mounted on ball bearings. Clutch drum is of cast iron, bolted to flywheel. Spring tension is supplied by two concentric coil springs of steel

Release collar carries ball thrust bearing, which is completely enclosed. Thrust of the clutch spring is taken on ball bearing on front end of transmission

Pilot ball bearing on front end of clutch shaft, completely enclosed by steel stamping and felt washers. Clutch brake is supplied

Propeller Shaft

This shaft is tubular, 2 in. diameter with $\frac{5}{32}$ in. walls. It is provided with two (2) universal joints. Slip is taken through 10-groove spline in the forward joint. Pins and bushings are hardened and ground steel. Joints are enclosed by metal boot in two sections, the outer section self-adjusting

Frames

Open-hearth, pressed steel, heat-treated. Height of frame from ground with load, approximately 27 in. Necessary clearances are obtained with straight side rails by tapering overall width of frame from 31 in. at front end to 34 in. at rear end. Length of frame back of toe board, 109 in. Overhang from center of rear axle to center of frame, 32 in.

Side members are $4\frac{1}{2}$ in. deep by $2\frac{1}{2}$ in. wide back of the driver's seat. From this point forward they taper both in width and depth to front end, where they are bent down and finished with drop forging for attaching front springs. From the point of greatest load back, the depth is constant, but the width tapers to $1\frac{3}{4}$ in. at the rear. The side members are reinforced by pressed-steel channel sections extending from the center cross member to a point ahead of the rear end of the front spring. These sections are pressed into the side members, riveted and welded

Cross members are pressed steel, three in number. Both front and center cross members have gusset plates stamped out integral with these members. Rear cross member is uniform section with the side members at this point and is riveted to them. A strong pressed-steel brace is diagonally across the rear corner

Pintle hook reinforcement. Because of the necessity for these chassis carrying pintle hooks for towing purposes, the frame is substantially reinforced at the rear by a system of diagonal braces, while the pintle hook proper is carried by a malleable iron guide which, together with the reinforcing members, is riveted to the rear cross member

Holes are drilled, not-punched

Brackets are jig drilled and machined or ground as per detail drawings, where they fit against the frame

Rivets are all driven hot

Attached to the front end of the frame is a bumper consisting of channel iron carried by means of recoil springs on malleable-iron brackets attached to each side member

Hood, Dash and Seat

The hood is of pressed steel with three hinge rods; center hinge rod is fastened at both ends. A woven lacing is used on both ends of the hood and a number of louvers are provided on each side of the hood. Pressed-steel hood sills close the gap between the lower edge of hood and top of frame

Dash is of wood covered on both sides with sheet metal, and carries only ignition switch, lamp bracket, oil gage and primer control. Floor and toe boards are of kiln-dried hard wood, oiled and finished, and are protected around the steering gear and pedal slots by steel plate

Radiator and Cooling System

Water capacity 3¾ gallons

Pump, centrifugal type, driven from magneto shaft

Radiator is composed of inner core with tanks and outer pressed steel cover which is removable and which rests directly on brackets riveted to the front cross member and is fastened to them by two (2) ½-in. studs screwed into steel reinforcements in bottom of case. Core is vertical, fin-tube type. Radiator outlet is provided with suitable drain cock at the lowest point of the water system and radiator is connected to the dash by means of a tie-rod swiveled at each end. Tie rod bracket on the radiator is riveted to the casing of the latter and has no connection with the tank

Fuel Tank and Fittings

Fuel tank is cylindrical in shape, has a capacity of 23 gallons, and is provided with filling hole of sufficient size to admit a man's arm for cleaning inside of tank. It is made of sheet steel with welded seams, double locked seams and soldered, or of terne plate. Ends of tanks are convex

Fuel line to the carbureter is of annealed copper tubing and is securely fastened where it touches the chassis frame.

Muffler and Exhaust

Exhaust pipe is of steel tubing

Muffler is built up of inner and outer shells with cast

heads held in place by long bolts. Muffler is supported by two malleable iron brackets to the frame

Lamps

Lamp equipment is as follows:

- 2 Corcoran No. 403 "prestolite" headlamps or their equivalent
- 2 Adams & Westlake No. 441 side lamps or their equivalent
- 1 Adams & Westlake No. 442 tail lamp or its equivalent

Side lamps are supported by iron bracket bolted to the dash. Tail lamp is protected from mud by sheet metal plate hung from the drawbar brace. This bar also serves as a license tag bracket

Prestolite tank will be furnished and pipe line therefrom shall be of brass tubing, on separate line to each headlamp, provided with some device so that either lamp may be disconnected without affecting the other. This pipe line shall extend at least 3 in. beyond radiator, and shall be properly secured wherever necessary

Fenders, Running Board, Etc.

Fenders are of heavy gage pressed steel in one piece and enameled all over. Front fenders are attached to sockets on frame by taper fittings and nuts. Steel aprons close space between fenders and chassis frame. (For ambulance purposes, front fenders only will be furnished)

Running boards are of kiln-dried hard wood, oil finished, bolted to pressed steel step hangers, protected on the edge by steel angles, which rest on top of the boards

Running boards must be extended on request and provided with suitable fasteners to take two standard ambulance boxes. This extension requires a third running board and step hanger on right-hand side.

Carbureter

1-in. automatic float feed

Throttle opening controlled by lever mounted on steering wheel quadrant and by foot accelerator

Priming device, now in use to facilitate starting in cold weather, is to be applied to all chassis and proper dash connections made therefor

Governor

Governor must be provided equivalent in action to Monarch 1-in. size, or such other device as may be approved by the Motor Transport Service

Ignition

Eisemann Model G-4, high tension, water-proof magneto, variable spark control lever on the steering wheel quadrant

Painting

Entire chassis is to be thoroughly cleaned of all dirt, grease and oil, and shall then be covered with one coat of suitable quality of paint of color to be selected by the Motor Transport Service. In shipment of metal parts not painted, they are to be covered with a rust resisting slush, and interior of the cylinders are to be protected by means of a small quantity of oil injected into each cylinder, after which the engine is turned over three complete revolutions by hand

CHRISTENING OF FIRST AMERICAN HANDLEY-PAGE

BEFORE an immense audience, composed of Government officials, Army and Navy officers of this country and of foreign countries, invited guests, and workers of the Standard Aircraft Corporation, the "Langley," the first American-built Handley-Page biplane bomber was duly "christened" on July 6, at the Standard plant, Elizabeth, N. J.

Dimensions and Equipment of the "Langley"

The Langley is of the so-called 1915 type of Handley-Page. The wing span is 150 ft., body length 63 ft.; its weight without bombs is 8360 lb. Its bomb-carrying capacity is from 3000 to 4000 lb.; the passenger capacity is 25, and it requires 60 gal. per hour of fuel. The official speed is 90 miles an hour, but it is capable of higher speed. Two Liberty engines are used. Two light Browning machine guns are mounted in the nose of the body, the engines being placed in armored compartments on either side.

Mrs. Harry B. Mingle, wife of the president of the Standard Aircraft Corporation, with the help of Colonel Simphill, the pilot on the official trip, smashed a bottle of champagne over the nose of the machine, and gave it the name Langley, in honor of the pioneer scientist in aeronautics. On the first trip, in addition to Lord Simphill, Major General William L. Kenly, U. S. A., chief of the Department of Military Aeronautics, was a passenger, as were also several others connected with the Royal Flying Corps.

Speakers at the Christening

Before and after the first flight, which lasted about half an hour, and on which an altitude of over 3000 ft. was reached, was made, a number of speeches were delivered from a stand set immediately adjacent to the machine. Among the speakers were Assistant Secretary of War Benedict Crowell; John D. Ryan, Director of Aircraft Production; Major General William Branker, representing the Air Minister of Great Britain; Sir Joseph Fowler, Deputy Director of Aircraft Production in Great Britain, and Senators J. S. Frelinghuysen of New Jersey and Charles S. Thomas of Colorado.

Secretary Crowell, the first speaker, expressed his regrets that Secretary of War Baker was unable to be present, and said that important matters concerning Russia had prevented his leaving Washington. He warned the people of this country of the danger of overconfidence, stating that the "Leviathan," formerly the German "Vaterland," was an example of overconfidence on the part of the Germans, who believed that we would never be able to use it because of its complicated construction. The Secretary complimented the working men and working women on their success in building the Langley.

Mr. Ryan asserted that the Langley is an earnest of what the United States is going to do in the future to promote its aircraft program. Thousands like it are under construction, all to be driven by the Liberty engine, the best and most powerful aeronautic engine ever built. Most of the dissatisfaction as to the production of aircraft, according to Mr. Ryan, has been caused by expectations beyond the possibility of performance. Continuing, he said:

"I can speak with knowledge on the subject, because I have for two months studied every cause of delay and

disappointment. I can speak freely, because no stretch of my conscience will let me claim any credit for what has been done up to the present day. My connection with the work of aircraft production for the Army has been so short that nothing I could have done or left undone could possibly have affected the accomplishment at this time.

"Much good work has been done by my predecessors, and I am taking this opportunity to assure the people of the country that in my opinion there has been no such delay with the work, or anything like such incapacity of those in charge, as has been indicated by some of the criticisms of the accomplishments, or lack of them, in production."

He then cited as examples of his statements the fact that the Liberty engine has been designed and put into production within fifteen months, and that during the months of May and June as many of these engines were built in our workshops as were produced of all types in the entire year of 1915 in Great Britain. Including other types of aircraft engines, there were more produced in our shops in the month of June than Great Britain produced in the whole year of 1915. Two of the best of foreign types of engines have been put into production here in quantity, and a third foreign engine of high power is in course of development. In closing, Mr. Ryan said:

"The facilities for the manufacture of aircraft engines and planes are being developed with all the speed and skill available. The fact that we have now entered the production stage gives assurance that the country and its Allies will soon realize the benefit of such a fighting air force as will satisfy our people and bring consternation and defeat to our enemy."

Professor Langley's Nephew a Speaker

John Langley, a nephew of Prof. Samuel P. Langley, was introduced, and expressed his gratification that the work done by his uncle in developing the scientific laws underlying the operation of aircraft had been recognized. He had come from Ann Arbor, Mich., where they were training hundreds of young Americans to fly, and they were ready to train just as many men as were needed for the machines that could be built.

General Kenly Represented Government

Mr. Mingle then presented the machine to General Kenly as the representative of the Government. In receiving it, General Kenly called attention to the fact that, contrary to what was sometimes thought, his department was ready to fly and fight all the machines that can be turned out in the United States. This statement was received with cheers and loud applause.

General Branker stated that people in England had been backing the Liberty engine for months, and assured the audience that the engine is a success. He said that even then he had a cablegram in his pocket asking that more of them be shipped before the end of this year. Sir Joseph Fowler also had a good word to say for the Liberty engine, and predicted that it would help to give the Allies an overwhelming superiority in the air. In closing the afternoon's program, patriotic speeches were made by Senators Frelinghuysen and Thomas of the United States Senate.

HEADLAMP TESTS IN NEW YORK STATE

A CONFERENCE was held recently in Albany, N. Y., attended by Secretary of State Francis M. Hugo, members of the Lighting Division of the S. A. E. Standards Committee and the Committee on Automobile Headlamps of the Illuminating Engineering Society, with representatives of manufacturers of glare eliminating devices. As a result, Secretary Hugo has issued the following instructions relating to headlamp tests.

SPECIFICATIONS FOR HEADLAMP TESTS

General Conditions of Acceptability—For the purpose of testing the intent of the New York State Law dealing with automobile headlamps, which provides that front lights shall be so arranged, adjusted and operated, as to avoid dangerous glare or dazzle, and so that no dangerous or dazzling light (projected to the left of the axis of the vehicle when measured 75 ft. or more ahead of the lamps) shall rise above 42 in. on the level surface on which the vehicle stands, such front lights as shall be sufficient to reveal any person, vehicle or substantial object on the road straight ahead of such motor vehicle for a distance of at least 200 ft., shall be deemed to comply with the law if the following conditions are fulfilled:

1 Any pair of headlamps under the conditions of use must produce a light which, when measured on a level surface on which the vehicle stands at a distance of 200 ft. directly in front of the car and at some point between the said level surface and a point 42 in. above this surface, shall be not less than 1200 apparent candlepower.

2 Any pair of headlamps under the conditions of use shall produce light which, when measured at a distance of 100 ft. directly in front of the car, and at a height of 60 in. above the level surface on which the vehicle stands, shall not exceed 2400 apparent candlepower, nor shall this value be exceeded at a greater height than 60 in.

3 Any pair of headlamps under the conditions of use shall produce a light which, when measured at a distance of 100 ft. ahead of the car, and 7 ft. or more to the left of the axis of the same, and at a height 60 in. or more about the level surface on which the vehicle stands, shall not exceed 800 apparent candlepower.

Conditions of Laboratory Test—In order to determine whether any particular device conforms to these requirements, it shall be subjected to laboratory tests according to the following specifications:

Number of Samples—Two pairs of samples of the device submitted shall be subjected to test. In the case of front glasses, the samples shall be of $9\frac{1}{4}$ in. diameter, when practicable.

REFLECTORS AND INCANDESCENT LAMPS

The reflectors used in connection with the laboratory tests shall be of standard high-grade manufacture, of 1.25-in. focal length, with clean and highly polished surfaces, and as nearly true paraboloidal form as is practicable, and as approved for this purpose by the National Bureau of Standards.

The incandescent lamps used in connection with the laboratory test shall be of standard high-grade manufacture and as approved for this purpose by the National Bureau of Standards.

Adjustments by Manufacturer's Representative—The manufacturer of the device shall be given due notice of

the date and place of test. Manufacturers' representatives present at the test shall be privileged to adjust their devices in any way which represents an ordinary and legitimate adjustment, including tilting the lamps or reflectors, which can be carried out by purchasers of the device, or such adjustment may be made by the laboratory expert acting on the instructions of the manufacturer. The character of the adjustment so made shall be carefully noted and stated in the report as "manufacturer's adjustment."

MAKING OF TESTS

The tests shall be as follows:

Test 1 Four-point test of pairs of samples.

A pair of testing reflectors, mounted similarly to the headlamps on a car, shall be set up in a dark room at a distance of not less than 60 ft. nor more than 100 ft. from a vertical white screen. If a testing distance of 100 ft. is taken, the reflectors shall be set 28 in. apart from center to center, and if a shorter testing distance is taken, the distance between reflectors shall be proportionately reduced. The axes of the lamps shall be parallel and horizontal, or as tilted in accordance with manufacturer's adjustment. The intensity of the combined light shall then be measured with each pair of samples in turn, the reflectors to be fitted with a pair of each of the following types of incandescent lamps, in turn:

1 Vacuum type, 6-8 volts, 17 mscp., G-12 bulb.

2 Gas filled type, 6-8 volts, 20 mscp., G-12 bulb.

The lamps shall be adjusted to give their rated candlepower. Measurements shall be made at the following points on the surface of the screen:

A In the median vertical plane parallel to the lamp axes, on a level with the lamps.

B In the same plane one degree of arc below the level of the lamps.

C In the same plane one degree of arc above the level of the lamps.

D Four degrees of arc to the left of this plane and one degree of arc above the level.

In an acceptable device both pairs of samples shall conform to the following specifications for observed apparent candlepower:

Points A and B The apparent candlepower shall not be less than 1200 at one of these points at least.

Point C The apparent candlepower shall not exceed 2400.

Point D The apparent candlepower shall not exceed 800.

Provided, however, that if the test indicates that a device which is unacceptable with either of the test lamps should come within the specifications with lamps of another candlepower or of the other type, the device may be passed with corresponding limitations as to the incandescent lamps to be used in connection with it.

Test 2 Complete Test of Single Sample.

A single sample taken as an average representative of the device as manufactured, shall be submitted to a complete test with a vacuum incandescent lamp of 17 cp., 6-8 volt rating in a G-12 bulb. This test shall show its light distribution characteristics by actual measurements made according to recognized and exact methods.

Distribution of Samples—One pair of the samples submitted shall be retained by the testing laboratory for

future reference and as samples of construction, and the other pair shall be returned to the office of the Secretary of State.

The report of the tests shall be rendered in duplicate to the Secretary of State, and shall be signed or initialled not only by the expert making the test, but also by an executive officer of the institution making the test.

It shall include a statement by the testing laboratory as to whether the device when properly applied substantially complies with Section 286 of the Highway Law, and shall suggest the maximum candlepower to be used with the same, and order the other conditions necessary in the operation of the device that it shall comply with the requirements of this specification.

REPORT OF JULY COUNCIL MEETING

A SPECIAL meeting was called on July 19 on the occasion of the dinner given to Sir Henry Fowler. Those present were: President C. F. Kettering; First Vice-president David Beecroft, Councilors B. B. Bachman and Charles M. Manly, Treasurer C. B. Whittelsey, and General Manager Coker F. Clarkson.

It was voted to transfer J. J. Amory, Glenn M. Ozias and Harry J. Porter from Associate to Member Grade, and to transfer R. G. Bradley from Junior to Member Grade.

Applications for a number applying for membership in the Society were gone over, these being assigned to grades as follows: 42 Members, 52 Associates, 20 Junior Members, 2 Student Enrollments, 2 Affiliate Members and 3 Affiliate Member Representatives.

It was voted to form a special committee, the members to be designated by the President, to investigate the status of alien enemies.

The Council decided to issue identification cards to all members in good standing. Treasurer Whittelsey and General Manager Clarkson will prepare a suitable form.

A. C. Bergmann was appointed a member of the House Committee to succeed the late Professor Frederick R. Hutton.

The following additions were made to the Standards Committee with assignment to the divisions indicated:

Aeronautic Division: D. L. Gallup, F. H. Trego, Noble Foss, Archibald Black, Alfred Verville, H. W. Christensen, and Glenn L. Martin. Conferees; Frank H. Russell, C. Voght and Maurice Olley.

Ball and Roller Bearings Division: R. E. Wells in place of J. G. Weiss.

Fuel and Lubrication Division: E. C. Newcomb, Lieut.-Col. J. G. Vincent. Conferees: Dr. E. W. Dean, Lieut.-Com. A. K. Atkins and Dr. H. C. Dickinson.

Miscellaneous Division: J. E. Genn in place of Berne Nadall.

Non-Ferrous Division: W. M. Corse as chairman.

Lighting Division: W. A. McKay. Conferee: Dr. C. H. Sharpe.

Motorcycle Division: Lieut. G. W. Herrington, as conferee.

The resignation of Dent Parrett as second vice-president and also as chairman of the Tractor Division was accepted with regret by the Council. Mr. Parrett has taken a commission as captain in the army.

It is expected that the next meeting of the Council will be held in October.

COUNCIL DINNER FOR SIR HENRY FOWLER

Methods by which engineering bodies interested in aeronautics in this country and in England can cooperate

to the best advantage were discussed at a dinner given July 19, in the Engineers Club, New York, in honor of Sir Henry Fowler, K. C. E., director of Aircraft Production in Great Britain. Sir Henry was the bearer of a letter from General Rugg, president of the Aeronautical Society of Great Britain, to the Society, this suggesting that Sir Henry take up with the Council matters of mutual interest.

A number of informal addresses were made at the dinner. The distinguished guest of the evening said that the S. A. E. and the Aeronautical Society of Great Britain could work together best by the interchange of technical papers, so that the members of one organization could keep in close touch with the activities of the other. President Kettering urged the necessity of doing now our utmost in constructive effort to further the aeronautic program. F. G. Diffin of the Bureau of Aircraft Production explained the necessity for international standardization of parts and materials, in view of the immense manufacturing program in contemplation. First Vice-president Beecroft expressed the opinion that the Allied nations should work together closely in all engineering design.

Second Vice-president Houston said that for the good of the great aeronautic industry we are now building up we should study more attentively the economic, social and industrial life of the allied countries. Councilor Charles M. Manly referred briefly to some of the work being done in developing aeronautic engines on the other side. The work now being done in establishing a uniform international aeronautical nomenclature was described by F. G. Ericson and General Manager Coker F. Clarkson.

Several speakers referred to the many courtesies shown by Sir Henry to the members of the American Aircraft Commission on its trip last winter to London. The dinner was therefore in part an acknowledgment of these courtesies; it was also a tribute to Sir Henry as the representative of the British Government and of the Aeronautical Society of Great Britain; it developed, however, into an emphatic expression of the feeling that for the good of humanity every step possible should be taken to interchange technical information and unify technical activities in the allied countries.

In addition to the speakers already mentioned there were present: Past-president James Hartness and Secretary Calvin W. Rice of the American Society of Mechanical Engineers; President-Elect Comfort A. Adams of the American Institute of Electrical Engineers; Councilors B. B. Bachman and Charles M. Manly, Treasurer C. B. Whittelsey, H. M. Swetland, N. B. Pope, Assistant Secretary Herbert Chase, Standards Manager M. W. Hanks and Field Secretary R. E. Plimpton.

PERSONAL NOTES OF THE MEMBERS

Stephen N. Bourne, formerly president and general manager, Bourne Magnetic Truck Co., Philadelphia, is now secretary, Emergency Fleet Corporation, 140 N. Broad St., Philadelphia.

Jerry W. De Cou, formerly factory manager, Smith Motor Truck Corp., Clearing, Ill., is now with Ross Gear & Tool Co., Lafayette, Ind.

W. N. Deisher, formerly service manager, The International Motor Co., Ottawa, Ont., is now general manager, Independent Auto Service, Ottawa, Ont.

William Harrower, formerly mechanical engineer, Brewster Co., Long Island City, is now mechanical engineer, designing special machinery, Warneke & Hay Co., Inc., 51-3 Maiden Lane, New York.

Jaul J. Palmer, formerly aeronautical engineer and draftsman, Aeromarine Plane & Motor Co., New York, is now chief aeronautical engineer, Christopher Hannevig, Inc., 32 Broadway, New York.

J. E. Roberts, formerly assistant sales manager of the Cole Motor Car Company, has been appointed general sales manager.

Arthur M. Rosenthal, plant engineer and master mechanic of the Champion Engineering Co. of Kenton, Ohio, is conducting a military night school for selective draft men. His work consists of training in automobile engineering as well as electrical engineering, and it is planned to take up other branches later.

Alex. Taub, formerly chief draftsman, Scripps-Booth Corp., Detroit, is now chief draftsman, Northway Motor Co., Detroit.

Joseph Tracy, consulting engineer, 1790 Broadway,

New York, has been appointed consulting engineer of the Bureau of Oil Conservation, United States Fuel Administration. Mr. Tracy was appointed by W. Champlin Robinson, Director of Oil Conservation. Mr. Tracy will conduct sanctioned tests of automobiles with reference to gasoline conservation. Automobiles of all types will be included in the investigations, the bureau, however, being particularly anxious to secure data on the road performance of motor trucks.

M. G. White, formerly chief inspector, aviation plant, Nordyke & Marmon Co., Indianapolis, is now sales engineer, U. S. Ball Bearing Mfg. Co., Chicago,

W. A. Hale, propeller expert of the Airplane Engineering Department, Bureau of Aircraft Production, McCook Field, Dayton, Ohio, was instantly killed during the fall of an airplane July 15, at the flying field of the Curtiss Aeroplane & Motor Corporation, Buffalo, N. Y., while performing his duty acting as an observer at a propeller test.

Mr. Hale was born in 1894. After graduating from Stevens Institute of Technology, he was employed for over a year at the Standard Aero Corporation, Elizabeth, N. J. Since July, 1917, Mr. Hale had been employed in airplane engineering work for the United States Signal Corps. He was first stationed in Washington, D. C., but was transferred to McCook Field in November, 1917. Mr. Hale was elected to Student Enrollment in the Society on May 29, 1916. Funeral services were held at his home in Cranford, N. J., on the 18th of July.

AIRPLANE MAIL SERVICE

THE importance attached to the establishment of the airplane mail service and the confidence felt in its development are shown by the resignation of Captain B. B. Lipsner of a commission in the aeronautic branch of the regular army to take charge of its operation, and by the action of the War Department in accepting his resignation for this purpose. The following information was kindly furnished by the Office of Information of the Post Office Department.

The service has gone far beyond the experimental stage; its practicability has been thoroughly established by the record made since its inauguration between Washington and New York on the 15th of May.

Captain Lipsner's surrender of an army commission to undertake this work is in anticipation of the very wide extension of the service with the termination of the war, when hundreds of airplanes and many aviators who have been in the military service will be available.

Captain Lipsner, a member of the society, is a specialist in automobile engineering and aeronautic and mechanical maintenance. He has made an extensive study of fuel and lubrication. He has been connected with the automobile industry for 18 years and is an inventor of several mechanical devices.

"I am looking forward to a rapid development in cross-country flying," Captain Lipsner said, "and this development will result only from a most carefully regulated and thorough mechanical maintenance system, the establishment of which will be given special attention under my supervision. Those employed in the service will be ex-

perienced men. Every one having anything to do with airplanes and equipment will be a specialist, and the habit of flying every day without hindrance by weather conditions will soon be acquired. The improvement and expansion of the service will result from careful analysis of every bit of information concerning the operation of the airplanes which will come from practical experience. Records made and carefully studied will serve as a guide to insure reliability and the covering of distances from point to point in the most direct way and in the shortest time. At the conclusion of the war I believe that the airplane mail service will be so thoroughly organized and established that its extension will be very rapid. A large number of aviators returning with the flying experience of war will undoubtedly find employment in this service, the efficiency of which will be still further enhanced by their practical skill in aerial navigation. The possibilities of the service are almost immeasurable, in my opinion, and incidentally with a practical carrying of mail will come scientific observations valuable to the Weather Bureau, of service to our coast defense, and a more complete topographical knowledge of the country. The familiarity with the conditions to be met in flying, as disclosed by topographical and weather observations, will result in constant improvement in the service. The next advance step in the operation of airplanes is the increase of engine life, and this I believe will be accomplished through the thorough maintenance system which will be applied in the daily service of carrying the mail."

Honor Roll of Society Members

THE following members have recently entered the services of the government in civilian or military capacities. This list, together with the "Service Directory of Members" following, is intended to contain the names of all members connected with the government, either in the military service or in civilian capacities. The names are listed in two parts, the first showing the members who have actually entered the military services, and the second those engaged as civilians. Every effort is made to have the addresses correct. It is therefore requested, in case of any error, that the member concerned immediately inform the New York office of the Society, so that a proper correction can be made. Members who have actually entered the service in any capacity, and who are not listed, should also write the details to the New York office.

MILITARY HONOR ROLL

Billings, C. M., first lieutenant, Ordnance Reserve Corps, U. S. A., Washington; Motor Equipment Section, Engineering Bureau, Office of Chief of Ordnance, 448 N. Capital Ave., Indianapolis.

Churchward, A. Gray, second lieutenant, Third Motor Mechanics Regiment, Signal Corps, A. E. F.

Dee, Simon R., private, Ordnance Motor Instruction School, Raritan Arsenal, Metuchen, N. J.

Gould, Allen A., captain, Inspection Section, Quartermaster Corps, N. A., Washington.

Kent, Richard, Company B, 302d Battery, Tank Corps, Camp Colt, Gettysburg, Pa.

Pagé, Victor W., captain, Aviation Section, Signal Corps, A. E. F., France.

Parrett, Dent, captain, Ordnance Department, U. S. A., Peoria, Ill., assigned to work of coordination of engineering, production and inspection in tractor factories in Middle West.

Seeley, S. Ward, private, 326th Field Signal Battalion,

Headquarters Det., Camp Wadsworth, S. C. (mail c/o C. E. Seeley, 4040 Green St., Philadelphia).

Sheahan, T. W., lieutenant, 12th Company, Third Training Battalion, Barracks 930, N. A., Camp Lee, Va.
Sturgis, G. B., chief machinist's mate, U. S. N. R. F., assistant inspector engineering material, Allis-Chalmers Co. Works, West Allis, Wis.

CIVILIAN HONOR ROLL

Brown, William G., Aircraft Division, Bureau of Construction and Repair, Navy Dept., Washington.

Claassen, Chas. W., inspector, Signal Corps, U. S. A., c/o The Engel Aircraft Co., Niles Ohio (mail) 2627 Highland Ave., Cincinnati, Ohio.

Hill, F. Leroy, Jr., aeronautical mechanical engineer, Production Engineering Dept., Signal Corps, U. S. A., 612 Lindsey Bldg., Dayton, Ohio.

King, Charles B., aeronautical mechanical engineer, Aviation Section, Signal Corps, c/o Duesenberg Motors Corp., Elizabeth, N. J.

Lincoln, C. W., aeronautical mechanical engineer, Bureau Aircraft Production, U. S. A., 1339 Newton St., N. W., Washington.

Mann, Arthur S., sanitary division, Medical Corps, N. A., Kankakee, Ill.

Munson, Charles C., production expert, Approvals Dept., Bureau of Aircraft Production, U. S. A., Washington.

Russel, A. W., Office of Assistant Secretary of War, War Dept., Washington.

Smith, Joseph E., mechanical draftsman, Engineering Division, Ordnance Bureau, Washington, (mail) Route 1, Box 169, Brazil, Ind.

Spencer, Leslie V., editor, technical publications, Airplane Engineering Dept., McCook Field, Dayton, Ohio.

Weekley, William G., aeronautical engineer, Engineering Dept., Curtiss Aeroplane & Motor Corp. (N. Elmwood Plant), Buffalo, N. Y.

Service Directory of Members

MILITARY SERVICE

ALDEN, HERBERT W., lieutenant-colonel, Motor Equipment Section, Carriage Division, Ordnance R. C., Washington.

ALDRIN, EDWIN E., lieutenant, Coast Artillery Corps, U. S. A., Ft. McKinley, Maine.

ALTER, ARTHUR S., chief machinist's mate, U. S. N. R. F., Washington.

AMON, CARL H., first lieutenant, Aviation Section, Signal R. C., 1st Motor Mechanics' Regiment, A. E. F., France.

ANDERSON, E. S., lieutenant, Aviation Section, Signal Corps, U. S. A., Rockwell Field, San Diego, Cal.

ANDERSON, OSCAR G., private, Co. A, 1st Prov. Ordnance Depot Battalion, U. S. A., (mail) P. O. No. 713, A. E. F.

ANDERSON, WILLIAM C., lieutenant, Engineer R. C., Brooklyn, N. Y.

ARNOLD, BION J., lieutenant colonel, Aviation Section, Signal R. C., Washington.

BAKER, FRANCIS H., chief machinist's mate, U. S. N. R. F., U. S. Naval Gas Engine School, Columbia University, New York.

BARE, ERWIN L., first lieutenant, Military Truck Production Section, Quartermaster Corps, U. S. N. A., 205 Union Station, Washington.

BLAKER, C., pilot cadet, Royal Flying Corps, Canada.

BARKER, C. NORMAN, pilot cadet, Royal Flying Corps, Camp Borden, Can.

BARNABY, R. S., ensign, U. S. N. R. F., Buffalo, assigned to inspection duty.

BARNES, NEVIN C., cadet, U. S. School of Military Aeronautics, Princeton, N. J.

BARTON, W. E., first lieutenant, Quartermaster R. C., Washington.

BATES, WM. O., JR., first lieutenant, Motor Equipment Section, Carriage Division, Ordnance R. C., Washington.

BEDFORD, E. A., U. S. A., A. E. F., France.

BENJAMIN, DAVID, Co. A, Training Detachment, Valparaiso, Ind.

BEVIN, SYDNEY B., captain, Engineering Bureau, Motor Equipment Section, Ordnance Department, U. S. R., Washington.

BIBB, JOHN T., JR., second lieutenant, reserve military aviator, Aviation Section, Signal R. C., Washington.

BIBB, JOHN T., JR., lieutenant, Aviation Section, S. R. C., Love Field, Dallas, Texas.

BIGELOW, A. C., second lieutenant, Motor Transport Section, Quartermaster Corps, U. S. A., 205 Union Station, Washington.

BILLINGS, C. M., first lieutenant, Ordnance Department, U. S. A., Rock Island, Ill.

BLAIR, C. A., corporal, 472d Aero Squadron, A. E. F., France.

BLANK, M. H., first lieutenant, Motor Equipment Division, Ordnance R. C., Grant Motor Car Co., Cleveland.

BLEAKLEY, P. A., sergeant, Ordnance Motor Instruction School, Holt Mfg. Co., Peoria, Ill.

BRINTON, BRADFORD, major, Quartermaster Corps, U. S. A., Washington.

BLOOD, HOWARD E., captain, Signal Corps, U. S. A., McCook Field, Dayton, Ohio, assigned as business executive, Airplane Engineering Department.

BOGGS, GEO. A., lieutenant, Quartermaster Corps, U. S. A.; (mail) Farmers Loan & Trust Co., Paris, France.

BOWEN, C. H., captain, Military Truck Production Section, Office of Quartermaster General, Washington.

BRANDMEIER, F. M., Signal Corps, U. S. A., Motor Transport School, Wilbur Wright Field, Fairfield, Ohio.

BRISCOE, FRANK, captain, Signal Corps, U. S. A., France.

BRITTEN, DANIEL L., captain, Ordnance R. C., Washington, assigned to Gun Division, Ordnance Section.

BRITTEN, WM. M., major, engineer of motor transportation, Quartermaster R. C., Washington.

BRODIE, JAMES S., Engineer Corps, U. S. N. A., Washington.

BROWN, HAROLD HASKELL, first lieutenant, Coast Artillery Corps, U. S. N. A., Fort Totten, N. Y.

BROWNE, ARTHUR B., major, Sanitary Corps, U. S. N. A., (mail) General Motors Co., Detroit.

CALLAN, JOHN LANSING, lieutenant, Reserve Flying Corps, U. S. N., U. S. S. Seattle, (mail) Postmaster, New York.

CAMPBELL, ARCHIBALD F., Aviation Section, Signal R. C., Washington.

CAMPBELL, LINDSEY F., 4th Battery, 2d P. T. R., Fort Sheridan, Ill.

CHASE, A. M., major, Ordnance Department, U. S. A., Washington.

- CLARK, EDWARD L., first lieutenant, Signal R. C., McCook Field, Dayton, Ohio.
- CLARK, ELMER J., captain, Signal R. C., Buffalo, N. Y.
- CLARK, VIRGINIUS E., lieutenant colonel, Signal Corps, U. S. A., Washington.
- CLARKE, A. FIELDER, Ground School, Aviation Section, U. S. N., Washington.
- CLEAVER, CHARLES F., captain, A. S. C., British War Department, London, Eng., assigned as inspector of mechanical transport, (mail) Peerless Motor Car Co., Cleveland.
- COCKRILL, EMMET, first lieutenant, Ordnance R. C., Ford Motor Co., Highland Park, Mich., assigned as production officer and mechanical engineer.
- COE, EDW. M., first lieutenant, Quartermaster Corps, U. S. A., Washington, (mail) Mechanical Repair Shops No. 302, A. E. F., France.
- COFFMAN, DON M., first lieutenant, Aviation Section, Signal R. C., Commercial Bldg., Dayton, Ohio.
- COLLINS, KENNETH G., first lieutenant, Signal R. C., A. E. F., Italy, (mail) 8th Aviation Instruction Center.
- COMSTOCK, HERBERT F., second lieutenant, Aviation Section, U. S. Air Service, A. P. O. 738, A. E. F., via New York.
- DAHLQUIST, CHAS. S., major, Quartermaster Department, U. S. N. A., Washington, assigned to Motors Division as supervisor of inspection on standardized military trucks.
- DAYTON, WILLIAM E., private, 306th Regiment, Field Artillery, U. S. N. A., Washington.
- DEEDS, EDWARD A., colonel, Equipment Division, Signal Corps, U. S. A., State, War and Navy Bldg., Washington.
- DE LA GARDE, LOUIS A. C., lieutenant, Motor Transport Army Service Corps, Chiswick, London, England, assigned as first-class engineer and workshops officer.
- DENISON, ARTHUR H., cadet, School of Military Aeronautics, Massachusetts Institute of Technology, Cambridge, Mass.
- DE LORENZI, ERNEST A., officer, Mechanical Transport, War Department, London, Eng.
- DE WITT, GEORGE W., ensign, U. S. N., France, (mail) U. S. S. Utowana, Postmaster, New York.
- DIAMOND, JAMES E., captain, Ordnance R. C., assigned to Motor Instruction School, Kenosha, Wis.
- DICKEY, HERBERT L., captain, Motor Equipment Section, Carriage Division, Ordnance R. C., Washington.
- DIMOND, G. A., first lieutenant, Motor Section, Ordnance R. C., Ft. Herring, Peoria, Ill.
- DONALDSON, FRANK A., captain, Carriage Division, Ordnance R. C., Sixth and B Sts., Washington.
- DOST, CHARLES O., first lieutenant, Aviation Section, Signal Corps, U. S. A., Ellington Field, Houston, Texas, assigned to engineering department.
- DU BOSE, GEO. W. P., major, American Ordnance Base Depot, A. E. F., France.
- DUNCAN, A. C., first lieutenant, Balloon Co. No. 7, Signal Corps, Aviation Section, Signal R. C., (mail) A. E. F., France.
- DUNTLEY, LLOYD B., first lieutenant, Ordnance R. C., Washington, assigned to Engineering Motor Equipment Section.
- EARLE, LAWRENCE H., captain, Ordnance R. C., assigned as inspector of ordnance, Holt Mfg. Co., Peoria, Ill.
- EELLS, PAUL W., lieutenant, 330th Field Artillery, Artillery R. C., Camp Custer, Battle Creek, Mich.
- EGGEN, O. E., 337th Field Artillery, Ordnance Corps, U. S. A., Camp Dodge, Iowa.
- EHLERS, PAUL, Battery E, 304th Field Artillery, U. S. A., A. E. F., France.
- ENGESER, BENJ. M., School of Military Aeronautics, Massachusetts Institute of Technology, Cambridge, Mass.
- ENGLISH, G. H., JR., first lieutenant, Ordnance R. C., Washington.
- EVANS, GORDON M., captain, Engineering Bureau, Motor Equipment Section, Ordnance Department, U. S. R., Washington.
- FARRELL, MATTHEW, captain, Quartermaster R. C., Washington.
- FINKENSTADT, EDWARD R., captain, Military Truck Production Section, Office of Quartermaster General, Washington.
- FISHLEIGH, W. T., major, Sanitary Corps, U. S. N. A., Washington, assigned as automobile engineer.
- FITZGERALD, GERALD, second lieutenant, Motor Truck Co. 348, Camp MacArthur, Texas.
- FLANIGAN, E. B., Officers' Reserve Training Camp, Plattsburg, N. Y.
- FORRER, J. D., captain, Engineer R. C., Washington.
- FOSS, CLARENCE M., captain, Ordnance R. C., Rock Island Arsenal, Rock Island, Ill., assigned to Motor Section.
- FOSTER, WILLIAM J., second lieutenant, Signal R. C., U. S. A., Washington, assigned to Engine Design Section, Airplane Engineering Department, Aviation Section.
- FOX, RUDOLPH H., first lieutenant, Ordnance R. C., Washington.
- FRANKLIN, G. KING, captain, Motor Section, Ordnance R. C., Washington.
- FREDRICKSEN, ARTHUR, first-class machinist's mate, Aviation Section, U. S. Navy, Camp Perry, Great Lakes, Ill.
- FREHSE, A. W., captain, Engineering Section, Motor Transport Service, Quartermaster Corps, N. A., Washington.
- FREVERT, CARL B., first lieutenant, Ordnance R. C., A. E. F., France.
- FRIEDGEN, A. E., first lieutenant, Engineering Section, Motor Transport Service, Quartermaster Corps, Washington.
- FULTON, RICHARD WALLACE, 5th Cadet Squadron, Aviation Section, Signal Corps, U. S. A., Houston, Texas.
- FURLOW, JAMES W., lieutenant colonel, Quartermaster Corps, U. S. A., Washington, assigned to Office of Quartermaster General.
- GAEBELEIN, ARNO W., lieutenant, Ordnance R. C., Washington, assigned to Carriage Division.
- GARDNER, LESTER D., captain, commanding officer, Aviation Section, Signal Corps Training Camp, Camp MacArthur, Tex.
- GETSCHMAN, G. F., second lieutenant, Ordnance R. C., Washington, (mail) Office of Inspector of Ordnance, Maxwell Motor Co., Chalmers Plant, Detroit.
- GEY, WILLIAM, 377th Truck Train, U. S. N. A., Camp Merritt, Tenafly, N. J.
- GEORGER, A. H., first lieutenant, Ordnance R. C., assigned as production officer, Maxwell Motor Co., Chalmers Motor Car Co., Detroit.
- GIBBS, S. E., U. S. School of Military Aeronautics, University of Illinois.
- GILLIS, HARRY A., major, Ordnance R. C., Washington.
- GLOVER, FRED., colonel, Quartermaster Corps, U. S. A., Washington, in charge of Army Motor Transport Service.
- GORRELL, EDGAR S., lieutenant colonel, Aviation Section, Signal Corps, U. S. A., Washington, (mail) Air Service, A. E. F., France.
- GRAHAM, LOUIS, captain, 309th Engineers, Engineers R. C., Camp Zachary Taylor, Ky.
- GRAY, B. D., major, chief of production engineering department, Equipment Division, Aviation Section, Signal Corps, U. S. A., Washington.
- GRAY, SAMUEL W., first lieutenant, Aviation Section, Signal R. C., U. S. A., (mail) 4th Co., 2d Motor Mechanics Regiment, Air Service, A. E. F., France.
- GREEN, GEO. A., major, Tank Section, British E. F., France.
- GUERNSEY, CHARLES, captain, Engineering Section, Motor Transport Service, Quartermaster Corps, U. S. N. A., Washington.
- GUTHRIE, JAMES, major, Ordnance R. C., Washington, assigned to Carriage Division, Engineering Bureau.
- HAESKE, F. C., lieutenant, U. S. A., Camp Sherman, Chillicothe, Ohio.
- HALL, C. M., major, Aviation Section, Signal Corps, U. S. A., Dayton, Ohio.
- HALL, ELBERT J., major, Engine Design Section, Engineering Division, Signal Corps, U. S. A., Washington.
- HALL, RICHARD H., JR., first lieutenant, Quartermaster Corps, U. S. N. A., Washington.
- HARMS, HENRY W., lieutenant colonel, Aviation Section, Signal Corps, U. S. A., (mail) Base Section No. 3, London, England.
- HARTMAN, A. A., private, U. S. N. A., Camp Devens, Ayer, Mass.
- HAWKE, CLARENCE E., Aviation Section, Signal Corps, U. S. A., Washington.
- HECKEL, C. E., second lieutenant, Motor Transport Division, Quartermaster Corps, Washington.
- HECOX, F. C., captain, Quartermaster Corps, U. S. A., Washington, assigned to Engineering Bureau, Motor Division, in charge of standardization of military motorcycles.
- HEGEMAN, HARRY A., major, Quartermaster Corps, U. S. A., Washington, assigned to office of Officer in Charge of Transportation.
- HENDERSON, S. W., first lieutenant, Ordnance R. C., Washington.
- HICKS, HARLIE H., captain, Aviation Section, Signal R. C., Washington.
- HOBBS, J. W., first lieutenant, Ordnance R. C., U. S. A., commanding officer, Fourth Heavy Mobile Ordnance Repair Shop, Camp Hancock, Ga.
- HOFFMAN, ROSCOE C., captain, Carriage Division, Motor Equipment Section, Ordnance R. C., Washington.
- HORINE, M. C., second lieutenant, Aviation Section, Signal R. C., Washington.
- HORNER, LEONARD S., major, Equipment Division, Signal Corps, U. S. A., Washington.
- HOWARD, WALTER S., first lieutenant, Military Truck Production Section, office of Quartermaster General, Washington.
- HOWLAND, WM. I. JR., lieutenant, Bureau of Ordnance, Navy Department, U. S. N. R. F., Washington.
- HOUSTON, HAROLD S., 3d Officers' Training Camp, Fort Monroe, Va.
- HOYT, F. R., lieutenant, Aviation Section, Signal R. C., A. E. F., France.
- HUBBELL, LINDLEY D., lieutenant colonel, U. S. N. A., Ordnance Department, Springfield, Mass., assigned as Officer in Charge, Hill Shops, Springfield Armory.
- HULL, M. LAIR, private, Ordnance Department, U. S. A., Washington, assigned to Trench Warfare Unit, Requirement Section, Control Bureau.
- JACO, E. L., captain, Engineer R. C., U. S. A., Washington, assigned to General Engineering Depots.
- JEFFREY, MAX L., first lieutenant, Military Truck Production Section, Office of Quartermaster General, Washington.
- JENKS, WESTON M., U. S. N. R. F., Massachusetts Institute of Technology, Cambridge, Mass., assigned as instructor in naval aviation.
- JENNINGS, J. J., first lieutenant, Engineer R. C., Office of Chief Engineer, A. E. F., P. O. 717.
- JONES, R. E., lieutenant, U. S. N. R. F., Washington, (mail) U. S. S. New York, Postmaster, New York City.
- JOY, HENRY B., lieutenant colonel, 4th Motor Mechanics Regiment, Signal Corps, U. S. A., Camp Hancock, Ga.
- JUNK, FRED H., second lieutenant, Aviation Section, Signal R. C., U. S. A., (mail) Signal Corps Aviation School, Carlstrom Field, Arcadia, Florida.
- KALE, LEWIS P., major, Quartermaster Corps, U. S. N. A., Washington.
- KENDRICK, JOHN F., Signal Corps, A. E. F., France, assigned to Research Inspection Division.
- KENNEDY, H. H., captain, Ordnance Department, N. A., Washington.

SERVICE DIRECTORY OF MEMBERS

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- KINKEAD, R. S., sergeant, Third Reserve Officers' Training Corps, Field Artillery, *Washington*.
- KLEMIN, ALEXANDER, sergeant, Signal Corps, U. S. A., McCook Field, *Dayton, Ohio*; assigned to research, Airplane Engineering Department, Aviation Section.
- KLINE, H. J., first lieutenant, Ordnance R. C., *Washington*, assigned to Anti-Aircraft Section, Carriage Division.
- KLOCKAU, W. F., private, Co. N, 4th Battalion, 163rd Depot Brigade, *Camp Dodge, Iowa*.
- KOHR, ROBERT F., second lieutenant, Engineers R. C., *Washington*.
- KOTTNAUER, EDWIN H., first lieutenant, inspector of ordnance, Ordnance R. C., U. S. A., (mail) Paige-Detroit Motor Car Co., *Detroit*.
- LONDON, CHARLES H., second lieutenant, Aviation Section, Signal R. C., *Camp Dick, Dallas, Tex.*
- LANE, ABBOTT A., first lieutenant, Aviation Section, Signal R. C., *Detroit, Mich.*
- LANZA, MANFRED, major, Quartermaster Corps, U. S. A., 303rd Motor Supply Train, *Camp Dix, Wrightstown, N. J.*
- LARSEN, LESTER REGINALD, second lieutenant, 107th Engineer Train, U. S. A., A. E. F., *France*.
- LAVERY, GEO. L., JR., first lieutenant, Ordnance R. C., *Washington*.
- LAY, ARTHUR J., captain, Aviation Section, Signal R. C., *Washington*.
- LE FEVRE, WM. G., lieutenant, Train, 77th Division, U. S. A., *Camp Upton, New York*.
- LEOPOLD, JOSEPH, second lieutenant, Aviation Section, Signal R. C., Massachusetts Institute of Technology, *Cambridge, Mass.*
- LEVY, ALFRED K., 3rd Ordnance Corps, assigned to Motor Equipment Section, U. S. N. A., *Washington*.
- LEWIS, CHARLES B., captain, Ordnance R. C., *Camp Lewis, American Lake, Wash*
- LEWIS, HARRY R., JR., captain, Ordnance R. C., *Springfield Armory, Springfield, Mass.*
- LIBBEY, E. B., captain, 102nd Ammunition Train, 27th Division, U. S. A., *Spartanburg, S. C.*
- LIPSNER, B. B., captain, Record Section, Aviation Section, Signal R. C., *Washington*.
- LOEB, S. ARTHUR, lieutenant, Signal R. C., U. S. A., (mail) General Motors Corp., Buick Division, *Flint, Mich.*
- LUDOLPH, F. E., Aviation Section, Signal Corps, U. S. A., Kelly Field No. 1, *S. San Antonio, Tex.*
- MACDONALD, K. B., lieutenant commander, U. S. N. R. F., Naval Aircraft Factory, League Island Navy Yard, *Philadelphia*.
- MCCORMICK, BRADLEY T., captain, Ordnance Department, U. S. A., *New York*.
- MCGILL, GEO. E., Equipment Division, Aviation Section, Signal Corps, U. S. A., Packard Motor Car Co., *Detroit*.
- MCINTYRE, H. C., captain, Ordnance R. C., *Washington*.
- McMURTRY, ALDEN L., captain, Quartermaster Corps, U. S. N. A., assigned to duty with War Plans Division, General Staff, War College, *Washington*.
- MACKIE, MITCHELL, major, Quartermaster Corps, U. S. A., A. E. F., *France*, assigned to Motor Truck Transport Section.
- MACCOULL, NEIL, JR., U. S. N. R., *Washington*.
- MARMON, HOWARD, major, Airplane Engineering Division, Signal R. C., McCook Field, *Dayton, Ohio*.
- MARSHALL, W. C., captain, Ordnance R. C., *Washington*.
- MARTIN, KINGSLEY G., captain, Quartermaster R. C., *Camp Dodge, Iowa*.
- MASON, GEO. R., lieutenant, A. E. F., *France*.
- MATTHEWS, MEREDITH, sergeant of ordnance, 7th Mobile Ordnance Repair Shop, 7th Division, U. S. A., (mail) *Camp McArthur, Waco, Texas*.
- MAY, HENRY, JR., first lieutenant, Quartermaster C., N. A., *Washington*, assigned as inspector of Type B engines.
- MAYER, JAMES L., lieutenant, 109th Engineers, *Camp Cody, New Mexico*.
- MEDER, CHARLES, 2nd Regiment, Co. 10, Section 1, Aviation Section, U. S. Naval Training Station, *Charleston, S. C.*
- MERCI, WILLIAM, Co. B, First Battalion, 153d Depot Brigade, *Camp Dix, Wrightstown, N. J.*
- METCALF, GEORGE R., JR., captain, Ordnance Department, U. S. N. A., *Washington*.
- MICHEL, C. A., The Naval Auxiliary, Reserve Officers' Training School, Steamer J. H. Sheadle, *Detroit River Station*.
- MIDDLETON, RAY T., first lieutenant, Air Service, A. E. F., *Paris, France*.
- MILLER, B. F., major, Quartermaster Corps, U. S. A., *Washington*.
- MILLER, C. A., first lieutenant, head checker, Quartermaster Corps, U. S. N. A., *Washington*.
- MILLER, DONALD G., first lieutenant, Ordnance R. C., U. S. A., (mail) Nash Motors Co., *Kenosha, Wis.*
- MITCHELL, C. B., lieutenant, 4th Motor Mechanics' Regiment, *Camp Hancock, Ga.*
- MOFFAT, ALEX. W., ensign, commanding U. S. S. "Tamarack" (S. P. 561), Naval Defense Reserve, Postmaster, Foreign Station, *New York*.
- MOFFETT, PAUL R., captain, Aviation Section, Signal Corps, *Washington*.
- MONCRIEFF, V. I., captain, Aviation Section, Signal R. C., *Washington*.
- MOORE, HAROLD T., sergeant, Quartermaster Mobile Repair Shop No. 302, Co. 2, American Expeditionary Forces, via *New York*.
- MORGAN, M. B., major, Engineering Bureau, Ordnance Department, *Washington*.
- MORRIS, PERCY G. B., Naval Aviation Corps, U. S. Navy, *Washington*, (mail) Aviation Headquarters, Great Lakes Naval Station, *Great Lakes, Ill.*
- MORSE, E. R., first lieutenant, Headquarters Sixth Mobile Ordnance Repair Shop, *Camp Wadsworth, S. C.*
- MURPHY, JOSEPH G., Sanitary Corps, U. S. N. A., *Washington*.
- MYERS, J. L., first lieutenant, Motor Equipment Section, Ordnance R. C., U. S. A., (mail) Allison Experimental Co., *Indianapolis*.
- NAHIKIAN, S. M., lieutenant, Aviation School, Massachusetts Institute of Technology, *Cambridge, Mass.*
- NICHOL, ALFRED H., Biplane Engineering Department, Signal Corps, U. S. A., *Dayton*.
- NORRIS, G. L., captain, Signal R. C., U. S. A., *Pittsburgh, Pa.*
- O'BRIEN, WM. B., JR., cadet, School of Military Aeronautics, Barracks No. 1, *Champaign, Ill.*
- OGBEN, CARL F., chief machinist's mate, Ordnance Inspection Department, U. S. N. R. F., *Washington*.
- OLDFIELD, LEE W., captain, Signal R. C., *Washington*, assigned as aeronautical engineer.
- OLIPHANT, LAURENCE, ensign, U. S. N., *Washington*.
- OMMUNDSON, H. P., chief quartermaster, Aviation, U. S. Naval Air Station, *Miami, Fla.*
- ONG, D. G., first lieutenant, Aviation Section, Signal Reserve Corps, U. S. A., *Dayton, Ohio*.
- ORTON, EDWARD, JR., major, Quartermaster R. C., *Washington*, assigned to Motor Transport Branch, Engineering Section.
- OSBORNE, ARTHUR D., second lieutenant, Ordnance R. C., U. S. A., *Washington*.
- OSWALT, W. L., first lieutenant, Ordnance R. C., 304th Mobile Ordnance Repair Shop, *Camp Meade, Md.*
- OTTO, HENRY S., lieutenant, Intelligence Section, A. E. F., *France*.
- PAGE, VICTOR W., first lieutenant, Aviation Section, Signal R. C., *Mincola, N. Y.*
- PAINE, C. L., captain, Ordnance R. C., U. S. A., Headquarters 7th Mobile Ordnance Repair Shop, *Camp McArthur, Waco, Texas*.
- PARKER, RICHARD E., captain, Quartermaster R. C., *Washington*, assigned to Southern Department.
- PARRAMORE, T. H., Motor Transport Service, Quartermaster Corps, U. S. A., *Washington*.
- PEARMAN, W. J., captain, Ordnance R. C., A. E. F., *France*.
- PECHNICK, FRANK J., 31st Balloon Co., U. S. A., Post Field, *Ft. Sill, Okla.*
- PETERSON, F. SOMERS, ensign, Naval Air Station, *San Diego, Cal.*
- PETTIS, JOHN G., chief machinist's mate, Aviation Section, U. S. N., *Washington*.
- PFEIFFER, BEN. S., first lieutenant, Ordnance R. C., Rock Island Arsenal, *Rock Island, Ill.*, assigned to Motor Section.
- PICKARD, LYNN W., chief machinist's mate, U. S. Naval Air Service, A. E. F., *France*.
- PIERCE, HUGH M., captain, Signal R. C., Call Field, *Wichita Falls, Texas*, assigned as engineer officer, Aviation Section.
- POST, EDWIN M., JR., lieutenant, U. S. Air Service, A. E. F., *France*.
- POTTER, AUSTIN E., lieutenant, U. S. N. R. F., U. S. Naval Aviation Forces, *France*.
- POWELL, W. B., captain, assigned as officer in charge of mechanical transport, Imperial Ministry of Munitions, *Quebec, Can.*, (mail) P. O. Box 194.
- PRATT, JESSE T., first lieutenant, Aviation Section, Signal Corps, U. S. A., *Washington*.
- PROCTOR, C. D., corporal, Ordnance Department, U. S. A., Ordnance Depot, *Chicago*.
- PULLEN, DANIEL D., lieutenant colonel, General Headquarters, A. E. F., A. P. O. 706, *France*.
- PURCELL, BERNARD A., captain, Quartermaster R. C., 307th Supply Train, *Camp Gordon, Ga.*, assigned as Commanding Officer.
- RANNEY, A. ELLIOT, major, Air Division, Signal Corps, U. S. A., *Washington*.
- RAWLEY, JOS., captain, Co. A, 310 Engineers, U. S. A., *Camp Custer, Battle Creek, Mich.*
- RICHARDSON, F. E., private, Engineer R. C., U. S. A., *Washington*.
- RIDDLE, E. C., cadet, Aviation Section, Gerstner Field, *Lake Charles, La.*
- RIFKIN, G., sergeant, inspector, Military Truck Production Division, Quartermaster Corps, U. S. N. A., (mail) Covert Gear Works, *Lockport, N. Y.*
- RITTER, E. R., first lieutenant, Ordnance R. C., U. S. A., *Washington*, assigned to Production Division, Carriage Section.
- ROBINSON, H. A., ensign, N. R., U. S. N., *Keyport, N. J.*
- ROSE, CHARLES B., major, chief of planes and engine inspection, Inspection Department, Signal Corps, U. S. A., *Washington*.
- ROMEYN, RADCLIFFE, Inspection Division, Ordnance Department, *Washington*.
- ROSENTHAL, WM. C., sergeant, Engineer O. T. C., *Camp Lee, Va.*
- ROUNDS, EDWARD W., U. S. N. R., U. S. Naval Aviation Detachment, *Cambridge, Mass.*
- RUMNEY, MASON P., captain, Production Division, Ordnance R. C., *Washington*.
- RUSSELL, EUGENE F., major, Ordnance Department, U. S. A., *Washington*.
- SANDT, A. R., sergeant, Ordnance Department, U. S. A., *Washington*, assigned to Motor Equipment Section, Engineering Bureau.
- SCHOENFUSS, F. H., captain, Gun Division, Production Section, Ordnance R. C., *Washington*.
- SCHOEFF, T. N., captain, Engineer R. C., *Washington*.
- SCHUFF, ARTHUR A., second lieutenant, Aviation Section, Engineering Department, New York Equipment District, Signal Corps, U. S. A., *Washington*.

SCOTT, ALLISON F. H., captain, Signal Corps, U. S. A., Langley Field, Hampton, Va., assigned to Aviation Section.

SELFRIDGE, S. W., first lieutenant, Ordnance R. C., Washington.

SEWALL, E. B., U. S. N. R. F., Washington.

SHAFFER, M. S., second lieutenant, Signal R. C., McCook Field, Dayton, Ohio, assigned to Airplane Eng. Div.

SKINNER, HARLAN C., private, Ordnance Department, U. S. A., Edgewood Arsenal, Edgewood, Md.

SLADE, ARTHUR J., captain, Aviation Section, Signal R. C., Washington.

SLOANE, JOHN E., first lieutenant, aeronautical engineer, Aviation Section, Signal Corps, U. S. A., Washington.

SMITH, EDSON H., ensign, U. S. N. R., (mail) American & British Mfg. Co., Bridgeport, Conn., assigned as assistant naval inspector of Ordnance.

SMITH, FRANK E., major, Signal Corps, U. S. A., Washington.

SMITH, G. W., JR., lieutenant, U. S. N. R., Naval Aircraft Factory, U. S. Navy Yard, Philadelphia.

SMITH, MARK A., first lieutenant, Marine Corps, U. S. N., Washington.

SMITH, WESTCOTT T., second lieutenant, Engineering Department, Aviation Section, Signal R. C., U. S. A., Chaunte Field, Rantoul, Ill.

SPENCE, HANS P., first lieutenant, Quartermaster Corps, U. S. A., Camp Custer, Mich.

SPERRY, LAWRENCE B., ensign, U. S. N. R. F., Massapequa, N. Y.

SPRAGUE, G. A., Co. D, 310th Engineers, Camp Custer, Battle Creek, Mich.

STAHL, R., lieutenant, U. S. Navy Seaplane Division, U. S. N. R., Washington.

STALB, A. ROLSTAN, ensign, U. S. N. R. F., (mail) Office of Operations, Aviation, Navy Annex, Washington.

STEINAU, J. M., sergeant, Sanitary Corps, U. S. N. A., Washington.

STEVENS, C. C., Motor Equipment Section, Ordnance Department, U. S. A., Washington, assigned as draftsman.

STRAHLMAN, OTTO E., first lieutenant, Aviation Section, Signal R. C., (mail) Mechanics Training School, Overland Bldg., St. Paul, Minn.

STRAUSS, M. FRANK, first lieutenant, 307th Mobile Ordnance Repair Shop, 82nd Division, U. S. A., Camp Gordon, Ga.

STREETER, ROBT. L., major, Ordnance Department, U. S. A., Rock Island Arsenal, Ill., in charge of truck and tractor experimental work.

STREICHER, GEORGE A., lieutenant, 3rd Engineers' Training Regiment, Camp Humphrey, Va.

SWEET, GEO. P., first lieutenant, Signal Corps, U. S. A., Washington, assigned to Aviation Section.

SWEET, GEO. W., captain, Ordnance Department, U. S. A., Washington, assigned as inspector of ordnance, Studebaker Corp., South Bend, Ind.

SWINTON, D. R., first lieutenant, Quartermaster Corps, U. S. N. A., Mobile Repair Shop 302, A. P. O. 708, A. E. F., via New York.

TAYLOR, PAUL B., sergeant, Medical Corps, U. S. A., Pontiac, Mich.

TAYLOR, S. G., JR., first lieutenant, Ordnance R. C., Washington, assigned to Ordnance Department.

TEETOR, D. C., captain, Ordnance R. C., Kenosha, Wis., assigned to Motor Section.

THOMPSON, H. E., first lieutenant, Motor Equipment Section, Carriage Division, Ordnance R. C., Washington.

THOMPSON, JOHN A., Ordnance Department, U. S. A., assigned to Engineering Bureau, Motor Equipment Section, Ford Bldg., Washington.

THOMPSON, W. H. F., Barracks 300, League Island Navy Yard, Philadelphia, assigned as machinist's mate.

TITSCH, WALTER H., captain, Quartermaster Corps, U. S. N. A., A. E. F., France.

TOLMAN, EDGAR BRONSON, JR., first lieutenant, 311th Engineers, U. S. A., Camp Grant, Rockford, Ill.

TURNER, HARRY C., captain, Engineer R. C., A. E. F., France.

TWACHTMAN, QUENTIN, first lieutenant, Engine Design Section, Signal R. C., Washington.

TYLER, P. O., captain, Ordnance R. C., U. S. A., Raritan Arsenal, Nison, N. J.

UNDERHILL, C. R., captain, radio officer, school for aerial observers, Aviation Section, Signal R. C., Langley Field, Hampton, Va.

VAIL, E. L., lieutenant, Aviation Section, Signal Corps, U. S. A., McCook Field, Dayton, Ohio, assigned as officer in charge of instruments and accessories.

VAN LOON, HENRY M., 310th Engineers, Camp Custer, Battle Creek, Mich.

VERITY, CALVIN W., captain, superintendent of forge shop, Ordnance R. C., Frankfort Arsenal, Philadelphia.

WADSWORTH, GEORGE R., major, chief engineer, Naval Aircraft Factory, Navy Yard, Philadelphia.

VINCENT, JESSE G., lieutenant colonel, U. S. A., assigned as commanding officer and chief engineer, McCook Field, Dayton, Ohio.

VONACHEN, F. J., lieutenant, Ordnance Department, U. S. N. A., Rock Island Arsenal, Rock Island, Ill.

WALDON, SIDNEY D., colonel, Equipment Division, Signal Corps, U. S. A., Washington.

WALL, WILLIAM GUY, lieutenant-colonel, Ordnance Department, U. S. A., Washington.

WALTER, MAURICE, first lieutenant, Ordnance R. C., Washington.

WALTON, FRANK, acting sergeant, Quartermaster Corps, U. S. A., Quartermaster Repair Unit, (mail) Washington, D. C.

WALTON, HAROLD E., 84th Aero Squadron, Signal Corps, U. S. A., Kelly Field, San Antonio, Texas.

WATSON, C. ROY, lieutenant, Aviation Section, Signal R. C., U. S. A., Washington.

WEEKS, PAUL, captain, Ordnance Department, U. S. A., Washington.

WEISS, ERWIN A., sergeant, Engineering Bureau, Motor Section, Ordnance Department, U. S. A., France, assigned as ordnance sergeant.

WELSH, W. E., Signal Corps, Aviation Section, U. S. A., Washington.

WETHERELL, S. P. JR., major, Quartermaster R. C., Motor Transport Service, A. E. F., France.

WHITTENEGER, OWEN M., first lieutenant, Ordnance R. C., Washington, assigned to Office of Chief of Ordnance.

WILSON, H. C., major, 58th U. S. Artillery, Coast Artillery Corps, Ft. Schuyler, N. Y.

WILSON, T. S., lieutenant colonel, Field Artillery, Santa Fe, N. M.

WODEHOUSE, B. A., sergeant, Co. A, 339th Infantry, Camp Custer, Mich.

WOLFF, RUDOLPH D., U. S. N. R. F. No. 5, Great Lakes, U. S. A., assigned as chief petty officer.

WOOD, C. G., first lieutenant, Quartermaster Corps, U. S. A., Washington, assigned to Motor Transport Section, Office of Quartermaster general.

WOOD, FRANK B., captain, technical expert, Air Division, Signal R. C., U. S. A., Washington, (mail) 3rd Motor Mechanics Regiment, Camp Greene, Charlotte, N. C.

WOOD, HAROLD F., lieutenant, Specification Section, Equipment Division, Signal R. C., Washington.

WOODS, S. H., captain, Military Truck Production Section, Office of Quartermaster General, Washington.

WORKMAN, LEE W., 670th Aero Squadron, Aviation Branch, Morrison, Va.

YONKIN, HARRY F., first lieutenant, Ordnance R. C., care Chief Ordnance Officer, A. P. O. No. 717, A. E. F., France, via New York.

CIVILIAN SERVICE

ADAMS, H. J., War Industries Board, Washington.

ADAMS, PORTER H., Office of the Section Commander, First Naval District, Rockford, Me.

ADAMS, RALPH L., Quartermaster Corps, U. S. A., Washington, assigned to Engineering Section, Motors Division.

AGINS, HERMAN J., draftsman, Quartermaster Corps, U. S. A., Washington, assigned to Motor Transport Division.

ANDERSON, E. S., mechanical engineer, Aviation Section, Signal Corps, U. S. A., Rockwell Field, N. Island, San Diego, Cal.

ANDERTON, H. C., aeronautical mechanical engineer, Production Engineering Department, Equipment Division, Signal Corps, U. S. A., Lindsey Building, Dayton, Ohio.

BARE, ERWIN L., automobile body designer, Office of Quartermaster General, Washington.

BARNABY, RALPH S., airplane inspector, Naval Reserve Flying Corps, Buffalo, N. Y.

BARNHARDT, GEO. E., aeronautical mechanical engineer, Signal Corps, U. S. A., Wilbur Wright Field, Dayton, Ohio, (mail) Cottage B6, Unit No. 1, Signal Corps, Aviation School, Fairfield, Ohio.

BARTON, CHAS. E., Signal Corps, U. S. A., McCook Field, Dayton, Ohio, assigned to Airplane Eng. Department.

BELLING, G. C. C., assistant inspector of engineering material, U. S. Navy, Buffalo, N. Y., (mail) Curtiss Aeroplane & Motor Corp.

BLAKEMORE, THOMAS L., aeronautical engineer, Bureau of Construction & Repair, U. S. N., Washington.

BOOTH, FRED C., draftsman, Motor Transport Division, Quartermaster Department, U. S. A., Washington, (mail) Room 205, Union Station.

BOURNE, STEPHEN N., secretary, Emergency Fleet Corporation, Philadelphia.

BOURQUIN, J. F., supervisor of chassis assembly, Military Truck Production Section, Office of Quartermaster General, Washington.

BRADFIELD, E. S., Engineering Department, Naval Factory, Philadelphia.

BREWER, ROBERT W. A., inspector of mechanical transports, British Army, London, Eng., (mail) Holt Mfg. Co., Stockton, Cal.

BROWN, WILLIAM G., Aircraft Division, Research Department, Bureau of Construction and Repair, Navy Department, Washington.

BUBNA, RICHARD C., designer, Engineering Office, Motor Transport Division, Quartermaster Corps, U. S. A., (mail) Room 305, Union Station, Washington.

BURTON, W. DEAN, aeronautical mechanical engineer, Signal Corps, U. S. A., Fort Omaha, Neb.

CALDWELL, FRANK W., aeronautical engineer, Aviation Section, Signal Corps, U. S. A., (mail) McCook Field, Dayton, Ohio.

CHAPMAN, ROBERT H., U. S. N., Spartanburg, S. C., assigned to Aeronautical Division.

CHAUVEAU, ROGER, aeronautical mechanical engineer, Aviation Section, Signal Corps, Washington.

CHERRY, RALPH E., Signal Corps, U. S. A., McCook Field, Dayton, Ohio, assigned to Airplane Engineering Department.

CLARK, ELMER J., Signal Corps, U. S. A., Portland, Ore., assigned as district manager of inspection.

CLARKE, THOMAS A., Signal Corps, U. S. A., Washington, assigned to Aviation Section as production expert.

CLEAVER, B. J., Medical Corps, U. S. A., General Motors Truck Plant, Pontiac, Mich.

COSTELLO, JOHN V., aeronautical engineer, airplane engineering division, Signal Corps, Dayton, Ohio.

COWAN, STUART, special work, Clothing and Equipment Division, Manufacturing Branch, Office of Quartermaster General, 109 East Sixteenth Street, New York.

SERVICE DIRECTORY OF MEMBERS

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- CORLETT, ROBERT C., student engineer of tests, U. S. Army Training School, Carnegie Institute of Technology, *Pittsburgh*.
- CREAGER, F. L., in charge of electrical equipment tests, engineering laboratory, Motor Transportation Section, Quartermaster Corps, U. S. N. A., (mail) Room 205, Union Station, *Washington*.
- CROW, HAROLD I., School of Military Aeronautics, University of California, *Berkeley, Cal.*, assigned as instructor in aeronautic engines.
- DAVIS, CHARLES ETHAN, consulting engineer, Ordnance Department, *Washington*.
- DEKLYN, JOHN H., technical assistant, National Advisory Committee on Aeronautics, *Washington*.
- DENKINGER, GEORGE MARSHALL, Signal Corps, U. S. A., *Washington*, assigned to Aviation Section as aeronautical engineer.
- DICK, ROBERT L., motor truck expert, Ordnance Department, *Camp Dodge, Iowa*.
- DIFFIN, F. G., assistant to Chief of Production Department, Aircraft Board, *Washington*.
- DRESSER, L. W., mechanical engineer, Motor Transportation Division, Engineering Section, Quartermaster Corps, U. S. A., (mail) Room 306, Union Station, *Washington*.
- DUCORRON, C. A. F., Signal Corps, U. S. A., care Nordyke & Marmion Co., *Indianapolis*, assigned as senior inspector, Signal Service at Large.
- DUVAL, EUGENE C., Signal Corps, U. S. A., assigned to Airplane Engineering Department, Mutual Home Bldg., *Dayton, Ohio*.
- EDGERTON, A. H., Signal Corps, U. S. A., 870 Woodward Ave., *Detroit*, assigned to Equipment Division as gage supervisor.
- EDMONDSON, D. E., U. S. Signal Service at Large, *Washington*, assigned as inspector of airplanes and airplane engines, Ericsson Mfg. Co., *Buffalo*.
- EISELE, WILLIAM S., draftsman, Aviation Section, Signal Corps, U. S. A., *Washington*.
- ELLIOTT, E. M., chief dispatcher, Emergency Fleet Corp., *Washington*.
- ERICSON, FRIEDHOF G., representative of Canada, International Aircraft Standards Board, *Washington*.
- FERRY, PHILLIPS B., Signal Corps, U. S. A., McCook Field, *Dayton, Ohio*.
- FLOGAUS, H. A., designer and checker, Motor Transport Engineering Division, U. S. A., (mail) Office of Quartermaster General, 205 Union Station, *Washington*.
- FOWLER, HARLAN D., aeronautical engineer, Production Engineering Division, Aviation Section, Signal Corps, *Washington*.
- FRENCH, H. J., senior inspector, Pittsburgh District, Aviation Section, Signal Corps, U. S. A., *Philadelphia*, assigned to Inspection Department.
- FROESCH, CHARLES, aeronautical mechanical engineer, Engineering Department, New York Equipment District, Aviation Section, Signal Corps, (mail) Aeronautical Engine Corp., *Long Island City, N. Y.*
- FRYER, ROY C., chief instructor, starting, lighting and ignition division, War Educational Department, University of Cincinnati, *Cincinnati*.
- GIBSON, HUGO, automotive purchasing, British War Mission, *New York*.
- GORMAN, E. J. B., U. S. Flying Corps, N. R. U. S. N., *Dayton, Ohio*, assigned to inspection of airplane engines, Dayton-Wright Aeroplane Co.
- GRIFFITH, LEIGH M., technical expert, National Advisory Committee for Aeronautics, 518 Munsey Bldg., *Washington*.
- GRIMES, C. P., Signal Corps, U. S. A., McCook Field, *Dayton, Ohio*, assigned to airplane engineering department.
- GUERNSEY, CHAS., chief draftsman, Quartermaster Corps, U. S. A., *Washington*, assigned to Engineering Section, Motors Division.
- HALE, W. A., aeronautical mechanical engineer, Signal Corps, U. S. A., *Dayton, Ohio*.
- HALLETT, GEO. E. A., aeronautical mechanical engineer, Aviation Section, Signal Corps, Arcade Bldg., *Washington*.
- HARRIGAN, F. P., aeronautical engineer, Equipment Division, Airplane Engineering Department, Signal Corps, U. S. A., McCook Field, *Dayton, Ohio*.
- HART, FRANK S., designer, Engineering Section, Motor Transportation Division, Quartermaster Corps, U. S. A., *Washington*.
- HECKEL, C. E., truck designer, Transport Division, Quartermaster Corps, U. S. A., *Washington*.
- HICKS, H. A., aeronautical engineer, Equipment Section, Aviation Section, Signal Corps, U. S. A., (mail) Major E. J. Hall, Lindsey Building, *Dayton, Ohio*.
- HOBBS, J. W., automobile expert, Ordnance Department, Rock Island Arsenal, *Rock Island, Ill.*
- HOLDEN, F. M., airplane engineering division, Signal Corps, U. S. A., *Washington*.
- HONIGMAN, JOS. K., instructor, U. S. School of Military Aeronautics Princeton University, *Princeton, N. J.*
- HOWER, HENRY M., production manager, New London Naval Base, *New London, Conn.*
- KING, CHARLES P., Aviation Section Signal Corps, U. S. A., *New York City*, assigned as aeronautical mechanical engineer.
- KINGSBURY, J. A., metallurgist, Aviation Section, Signal Corps, U. S. A., (mail) Trego Motors Corp., *New Haven, Conn.*
- KISHLINE, FLOYD F., laboratory assistant, Quartermaster Corps, *Washington*.
- KROEGER, F. C., Quartermaster Corps, U. S. A., *Washington*, assigned as engineer on electrical equipment.
- KUMPEL, REUBEN, U. S. N., Naval Air Station, *Pensacola, Fla.*, assigned to Bureau of Steam Engineering.
- LADDON, I. M., aeronautical engineer, Signal Corps, U. S. A., McCook Field, *Dayton, Ohio*.
- LANE ABBOTT A., inspector, Aviation Section, Signal Corps, (mail) Room 52, 870 Woodward Avenue, *Detroit*.
- LEAVELL, R. A., Federal Board for Vocational Education, *Washington*, assigned as associate professor of mechanical engineering in charge of automobile instruction, Camp Joseph E. Johnston, *Jacksonville, Fla.*
- LINCOLN, C. W., aeronautical engineer, airplane engine department, Equipment Division, Signal Corps, U. S. A., *Washington*.
- LONGLETT, WESLEY, Signal Corps, U. S. A., assigned as inspector on airplane engines at The Nordyke & Marmion Co., *Indianapolis*.
- LOOMIS, ALLEN, Bureau of Aircraft Production, McCook Field, *Dayton*, assigned to airplane engineering department.
- LOUDON, WARREN P., expert cost accountant, Signal Corps, U. S. A., *New York*.
- MACPHERSON, JAMES W., inspector of airplanes and airplane engines, Signal Service at Large, Signal Service, U. S. A., *Washington*.
- MACDONALD, K. B., consulting engineer, Naval Aircraft Factory, *League Island, Philadelphia*.
- MCDONALD, E. G., aeronautical engineer, Airplane Engine Division, Signal Corps, U. S. A., McCook Field, *Dayton, Ohio*.
- MCCAIN, GEO. L., Signal Corps, U. S. A., *Dayton, Ohio*, assigned to airplane engineering department, Engine Design Section.
- MCMASTER, MARCENUS D., Aviation Section, U. S. N. R. F., *Washington*, assigned as warrant officer and machinist.
- MENNEN, F. E., Quartermaster Corps, U. S. A., *Washington*, assigned to Transportation Division.
- MILLAR, THOMAS H., JR., Engineering Section, Motors Division, Quartermaster Corps, *Washington*, (mail) Office of Quartermaster General.
- MILLER, C. S., automotive engineer, U. S. A., *Washington*; (mail) 1750 Elkin Ave., *New Albany, Ind.*
- MOORHOUSE, A., Signal Corps, U. S. A., Lindsey Bldg., *Dayton, Ohio*, assigned as engineer in Airplane Eng. Dept.
- MORGAN, G. W., supervisor of plant survey, Military Truck Production Section, Office of Quartermaster General, *Washington*.
- NELSON, A. L., aeronautical engineer, Airplane Engineering Department, Aviation Section, Signal Corps, U. S. A., McCook Field, *Dayton, Ohio*.
- NEUMANN, JOHN W., Planning Section, Machine Division, U. S. Navy Yard, *Philadelphia*.
- O'MALLEY, JOHN M., aeronautical engineer, Aviation Section, Signal Corps, U. S. A., Rockwell Field, *San Diego, Cal.*
- OTIS, J. HAWLEY, Ordnance Department, U. S. A., Camp Dodge, *Des Moines, Iowa*.
- PARISH, W. F., Supply Section, Department of Military Aeronautics, *Washington*, assigned as chief of Oil and Lubrication Branch.
- PARKER, VICTOR C., Signal Corps, U. S. A., *Washington*, assigned to Equipment Division.
- PARRIS, JR., EDWARD L., senior inspector, Aviation Section, Signal Corps, (mail) Ericsson Mfg. Co., *Buffalo*.
- FERRIN, J. G., aeronautical mechanical engineer, Production Engineering Department, Signal Service at Large, care Major B. D. Gray, *Washington*.
- POLLOCK, RAY C., Signal Corps, U. S. A., *Buffalo*, assigned as airplane engine inspector.
- PROCTOR, C. D., Ordnance Department, U. S. A., Rock Island Arsenal, *Rock Island, Ill.*, assigned to Motor Section, Carriage Division.
- RANDALL, J. M., assistant inspector of ordnance, Ordnance Department, U. S. A., (mail) Nash Motors Co., *Kenosha, Wis.*
- REID, JAMES, Boston Navy Yard, Bldg. 105, Shipsmith's Office, *Boston*.
- RICE, HARVEY M., inspector, Signal Service at Large, Signal Corps, (mail) Willys-Overland Co., *Toledo, Ohio*.
- RIDER, W. KEITH, Production Engineering Department, Signal Corps, U. S. A., *Dayton*.
- RIPPINGILLE, E. V., Aviation Section, Signal Corps, *Washington*.
- ROBERTS, D. S., Signal Corps, U. S. A., care Sturtevant Aeroplane Co., *Boston*, assigned as inspector of airplanes.
- ROBERTS, SAMUEL B., Signal Corps, U. S. A., care Sturtevant Aeroplane Co., *Boston*, assigned as inspector of airplanes.
- RUCKSTELL, G. E., Signal Corps, U. S. A., assigned as aeronautical mechanical engineer, *Detroit*.
- RUSSELL, L. L., Engineering Office, Motors Division, U. S. A., (mail) Office of Quartermaster General, 305 Union Station, *Washington*.
- RYMARCZICK, GUSTAV M., Signal Corps, U. S. A., (mail) Splittorf Electrical Co., *Newark, N. J.*, assigned to Aviation Sect., as senior inspector, Signal Service at Large.
- SALISBURY, EDWARD V., chief of motor transportation, American International Corp., Government Shipbuilding Yard, Hog Island, *Philadelphia*.
- SHAW, B. RUSSELL, Aviation Engineering Department, Signal Corps, U. S. A., McCook Field, *Dayton, Ohio*.
- SCHELL, JOHN A., aeronautical mechanical engineer, Signal Corps, U. S. A., McCook Field, *Dayton, Ohio*.
- SEABURY, W. M., Field Hospital, No. 337, Camp Custer, *Battle Creek, Mich.*
- SEABURY, W. WARNER, Signal Corps, U. S. A., Bureau of Standards, *Washington*, assigned to testing of aviation instruments.
- SEARLE, C. A., auto-parts inspector, U. S. A., *Washington*.
- SELLERS, MATTHEW B., Naval Consulting Board, *New York*.
- SERRELL, ERNEST, aeronautical mechanical engineer, Aviation Section, Signal Corps, *Washington*.
- SHILLINGER, G. P., Ground Officers' Engineering School, Kelly Field No. 1, *San Antonio, Tex.*, assigned as instructor in ignition, starting and lighting.
- SIMPSON, HOWARD W., Signal Corps, U. S. A., *Detroit*, assigned as inspector of aircraft engines, Inspection Section, Equipment Division, (mail) 870 Woodward Ave.
- SOULIS, WILBUR T., mechanical engineer, United States Gas Defense Plant, *Long Island City, N. Y.*

SPRAGLE, R. L., inspector, Detroit district, Signal Corps, U. S. A., Garfield Building, *Detroit*.
 STANTON, D. T., military instructor, U. S. Army School of Military Aeronautics, Cornell University, *Ithaca, N. Y.*
 STEARNS, L. C., technical assistant, National Advisory Committee for Aeronautics, *Washington*.
 STOUT, WILLIAM B., technical adviser, Aircraft Board, *Washington*.
 STUART, H. R., Signal Corps, U. S. A., Lindsey Building, *Dayton, Ohio*, assigned as aeronautical mechanical engineer, Production Engineering Department.
 SUTTILL, ALBERT G., Merchant Shipbuilding Corp., *Bristol, Pa.*, assigned as inspector of machinery, Boston District; (mail) General Delivery, *Hyde Park, Mass.*
 THIBAUT, F. J., aeronautical mechanical engineer, Signal Corps, U. S. A., McCook Field, *Dayton, Ohio*.
 THOMAS, T. R., mechanical engineer, Signal Corps, U. S. A., McCook Field, *Dayton, Ohio*.
 TONE, FRED I., inspector, Aviation Section, Signal Corps, *Washington*.
 TRACY, PERCY WHEELER, supervisor of parts plants, Military Truck Production Section, Office of Quartermaster General, *Washington*.
 UTZ, JOHN G., supervisor of inspection, Office of Military Truck Production Section, Office of Quartermaster General, *Washington*.
 VOHRER, W. R., draftsman, Engineering Section, Motor Division, Quartermaster Corps, U. S. A., *Washington*.
 WADE, GUSTAV, inspector, Aviation Section, Signal Corps, *Dayton, Ohio*.

WALDON, C. O., National Bureau of Standards, *Washington*, assigned as laboratory assistant, Military Research Gas Engines.
 WALDRON, RUSSELL E., Signal Corps, U. S. A., *Detroit*, assigned to Equipment Division.
 WALKER, KARL F., automotive engineer, Quartermaster Corps, U. S. A., *Washington*, assigned to Engineering Laboratory.
 WALTER, JOHN M., mechanical draftsman, Bureau of Ordnance, Navy Department, *Washington*.
 WARNER, EDWARD P., aeronautical engineer, Signal Service at Large, U. S. A., Mass. Institute of Technology, *Cambridge, Mass.*
 WATERHOUSE, W. J., aeronautical engineer, Aviation Section, Signal Corps, (mail) Dayton-Wright Airplane Co., *Dayton, Ohio*.
 WEAVER, E. W., aeronautical engineer, Engineering Department, Naval Aircraft Factory, Navy Yard, *Philadelphia*.
 WHINNE, WILBUR H., inspector, Quartermaster Corps, U. S. A., *Detroit*.
 WILLIAMS, S. T., Naval Aircraft Factory, Navy Yard, *Philadelphia, Pa.*, assigned as aeronautical mechanical engineer in Engineering Department.
 WINTER, E. A., War Department, Rock Island Arsenal, *Rock Island, Ill.*
 WOODWORTH, P. B., district educational director, War Department, Tribune Building, *Chicago*, (mail) 5809 Race Ave., *Chicago*.
 WORTHEN, C. B., inspector, Aviation Section, Signal Corps, U. S. A., *Washington*.
 WRIGHT, E. H., Bureau of Aircraft, McCook Field, *Dayton*, assigned to factory manager's office, experimental department.
 YOUNGER, JOHN, Quartermaster Corps, U. S. A., *Washington*, assigned to Motor Transport Service, as chief engineer.

Applications for Membership

The applications for membership received between July 2 and July 25, 1918, are given below. The members of the society are urged to send any pertinent information with regard to these names which the Council should have for consideration prior to their election. It is requested that such communications from members should be sent promptly.

ADAMS, MERRITT H., works manager, engineer, Vim Motor Truck Co., *Philadelphia*.
 ALKIRE, W. D., owner, manager, Eureka Auto Shop, *Chicago*.
 ANDERSON, CARL LUDWIG, superintendent, Mechanics Machine Co., *Rockford, Ill.*
 ATWOOD, HARRY N., 2nd vice-president, Carolina Aircraft Co., *Raleigh, N. C.*
 BARBOUR, ROBERT, vice-president, Parrett Tractor Co., *Paterson, N. J.*
 BARTHOLOMEW, J. E., president, Avery Company, *Peoria, Ill.*
 BATES, A. H., chief engineer, Emerson-Brantingham Co., *Minneapolis*.
 BAYLOR, C. A., superintendent, Great Western Mfg. Co., *Laporte, Ind.*
 BEEDE, RAY L., vice-president, manager, Southern Motor Co., *Ardmore, Okla.*
 BECKMAN, WILLIAM R., experimental engineer, designer, Duesenberg Motors Corp., *Elizabeth, N. J.*
 BREAKER, H. O., general manager American Incandescent Heat Co., Inc., *Boston*.
 BROWN, FRED., president, engineer, Brown & Kragness, *New York City*.
 BUERKLE, LEWIS J., electrical apparatus engineer, Dayton Engineering Laboratories Co., *Dayton*.
 BUTZOW, L. J., designing engineer, Remy Electric Co., *Detroit*.
 CAWTHRA, ERNEST H., production engineer, Curtiss Aeroplane & Motor Corp., *Buffalo*.
 COPE, LORENZO S., metallurgist, Hoover Steel Ball Co., *Ann Arbor, Mich.*

DAVIS, CHARLES W., chief engineer, The Torrington Co., *Torrington, Conn.*
 DICKINSON, DR. H. C., physicist, Bureau of Standards, *Washington*.
 DICKOVER, I. C., service manager, Service Motor Truck Co., *Wabash, Ind.*
 DORFMULLER, ANTON C., engineer, S. K. F. Ball Bearing Co., *Hartford, Conn.*
 GANNETT, HERBERT I., general manager, chief engineer, The Douglas Motors Corp., *Omaha, Neb.*
 GRAHAM, R. C., chief draftsman, Ross Gear & Tool Co., *LaFayette, Ind.*
 GRAMBSCH, R. H., designer, Dayton Engineering Laboratories Co., *Dayton*.
 HAHN, E. A., asst. chief engineer, Great Motor Car Corp., *Cleveland*.
 HEPNER, ALFRED K., general manager, Bearing Service Company, *Detroit*.
 HIMES, WALTER H., chief engineer, Bessemer Motor Truck Co., *Grove City, Pa.*
 HOBBS, J. O., JR., treasurer, American Locomotive Co., *New York*.
 KENNEY, W. C., manager of mechanical department, Hyatt Roller Bearing Co., *Detroit*.
 KELLEY, W. H., service manager, Winther Truck Co., of N. Y., *New York City*.
 KIMBALL, H. G., patent lawyer, Wetmore & Jamner, *New York City*.
 KIMMEL, ALFRED W., aeronautical mechanical engineer, Bureau of Aircraft Production, U. S. A., McCook Field, *Dayton*.
 KRAGNESS, EDWARD O., treasurer, engineer, Brown & Kragness, *New York City*.
 LANSING, J. TWICHELL, second assistant manager, Bijur Motor Appliance Co., *Hoboken, N. J.*
 LORENZ, HAROLD V., draftsman, Twin City Four Wheel Drive Co., *St. Paul*.
 MCNAMARA, J. H., factory manager, Curtiss Aeroplane & Motor Corp., *Hammondsport, N. Y.*
 MACK, STANLEY, tool inspector, Pierce-Arrow Motor Car Co., *Buffalo*.
 MARTINDALE, THOMAS B., authorized agent for Ford cars, *Philadelphia*.
 MIDGLEY, THOMAS, JR., mechanical engineer, The Dayton Metal Products Co., *Dayton*.
 MILLMANN, JOSEPH C., president, Titan Truck Co., *Milwaukee*.
 MOBLEY, C. S., automotive engineer, Motor Transport Div., Quartermaster General Office, *Washington*.
 NICHOLS, L. W., secretary, The Gary Motor Truck Co., *Gary, Ind.*
 PIERSON, TORVALD, body designer, Republic Motor Truck Co., Inc., *Alma, Mich.*
 RODDEWIG, GIL F., draftsman, Velie Motors Corp., *Moline, Ill.*
 SCHAFER, JOHN V., assistant engineer, Caskey-Dupree Mfg. Co., *Marietta, Ohio*.
 SCHRODER, FRED., general factory superintendent, The Stewart Mfg. Corp., *Chicago*.

APPLICATIONS FOR MEMBERSHIP

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SILVA, ERNEST H., president, general manager, The E. H. Silva Co., *Covington, Ky.*; consultant, The Dayton Storage Battery Co., *Dayton*; consultant, The Auto Electrical Supply Co., *Columbus, Ohio*; instructor, War Education Dept., *Cincinnati University, Cincinnati*.

SILVERMAN, JACOB HAROLD, designer, layoutman, Chevrolet Motor Car Co., *New York*.

SIMMONS, C. F., factory manager, Airplane Engineering Department, McCook Field, *Dayton*.

SPERRY, ROBERT O., salesman, American Bronze Corp., *Berwyn, Pa.*

SPONG, NORMAN C., designer, International Harvester Corp., *Akron, Ohio*.

STETTER, JOHN M., president, Muncie Cap & Screw Co., *Muncie, Ind.*

TABER, PERCY EDWARD, assistant manager, director, Taber-Bigelow Co., Inc., *San Francisco*.

TAYLOR, KENNETH S., chief engineer, Ericsson Mfg. Co., *Buffalo*.

VIAL, ETHAN, managing editor, *American Machinist*, *New York City*.

WAGNER, P. C., sales engineer, Blood Brothers Machine Co., *Allegan, Mich.*

WENDLAND, H. J., supervisor of materials and inspector, Dorris Motor Car Co., *St. Louis*.

WEMP, E. E., chief engineer, Denby Motor Truck Co., *Detroit*.

WILSON, EDGAR HUNT, president, general manager, Dural Rubber Corp., *Flemington, N. J.*

WOOD, LLOYD E., assistant chief engineer, Mitchell Motors Co., *Racine, Wis.*

WRIGHT, DANIEL M., secretary, treasurer, general manager, The Henry Wright Mfg. Co., *Hartford, Conn.*

Applicants Qualified

The following applicants have qualified for admission to the Society between June 24 and July 17, 1918. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff.) Affiliate; (Aff. Rep.) Affiliate Representative; (S. E.) Student Enrollment.

CHAMBERLAIN, WILLIAM A. (M.) chief of technical staff (American) Handley-Page Co., *London*, and Standard Aircraft Corp., *Elizabeth, N. J.* (mail) 248 So. Broad St., *Elizabeth, N. J.*

DAUM, GEORGE W. (A) general superintendent, Pennsylvania Rubber Co., *Jeannette, Pa.*

FITNESS, R. J. (M) assistant chief engineer, Pan Motor Co., *St. Cloud, Minn.*

GIBBS, S. E. (J) instructor in Motor Dept., School of Military Aeronautics, University of Illinois, (mail) 402 E. White St., *Champaign, Ill.*

HESS, SAMUEL P. (M) assistant manager, Spring Department, Detroit Steel Products Co., 119 Marston Ave., *Detroit*.

KLINGER, P. W. (M) chief engineer, Dayton Steel Foundry Co., *Dayton*.

LOVE, JOHN E. (A) sales agent, 1156 Penobscot Bldg., *Detroit*.

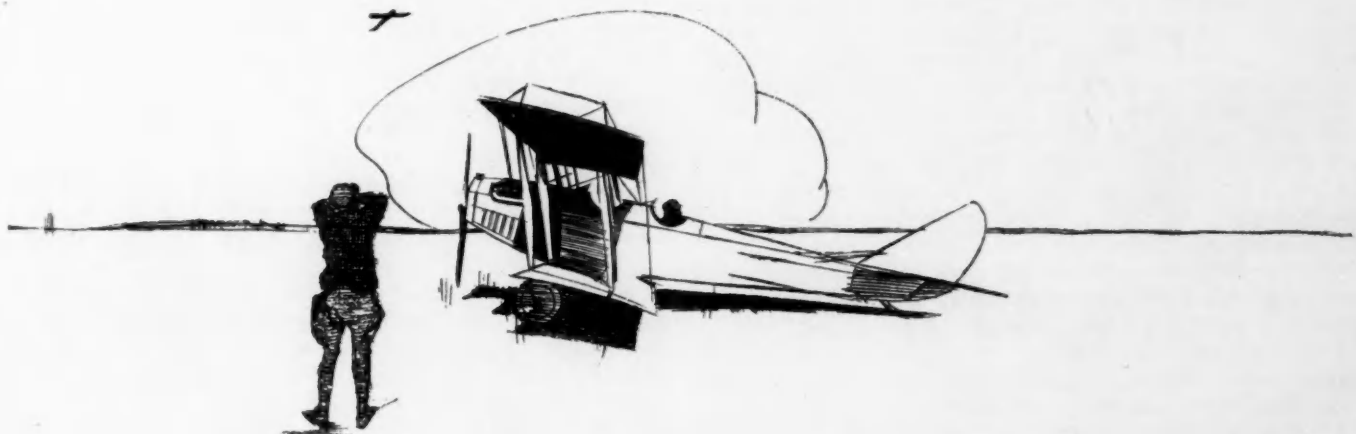
ORROK, GEORGE A. (M.) consulting engineer, 17 Battery Pl., *New York City*, (mail) R. F. D. No. 2, *Willimantic, Conn.*

POLLOCK, ROBERT T. (M.) president and treasurer, Robert T. Pollock Co., 68 Devonshire St., *Boston, Mass.*

SHIMO, HAYES G. (A) assistant manager, Bearings Co. of America, *Lansing, Mich.*

WHITE, GEORGE A. (M) mechanical engineer, The Sparks-Withington Co., *Jackson, Mich.*

GIO ANSALDO & Co. (Aff.) 80 Maiden Lane, *N. Y. City*. Representative Sebastiano Raimondo.



Book Reviews for S. A. E. Members

This section of THE JOURNAL contains notices of the technical books considered to be of interest to members of the Society. Such books will be described as soon as possible after their receipt, the purpose being to show the general nature of their contents and to give an estimate of their value.

THE AEROPLANE SPEAKS. By H. Barber. Published by McBride, Nast & Co., New York. Cloth, 6½ by 10 in., 144 pp., 36 Plates. Price \$3.

In England Captain Barber's book has gone through five editions in less than a year. Over there it is undoubtedly the most popular of the large number of books dealing with the elementary principles of flight. The author, a captain in the Royal Flying Corps, has had long experience in designing, building and flying aircraft.

The book is divided into two parts, the first, a prologue in which the airplane really does speak, as the author makes Efficiency, Propeller, Stability and all the other different principles involved in aircraft design and operation, tell their story in person. The result is a most entertaining narrative in which, after the elementary principles are explained, the Designer appears on the scene and the machine is completed. Of course, a test is necessary at this point, and finally a cross-country run is made, which gives the author an opportunity to describe how the pilot follows his course, how he makes an emergency landing, and finally how he returns to the home aerodrome after a successful flight, to gladden the heart of the Designer, who walks off arm in arm with Efficiency, who means to him, according to the author, the passion of speed, the lure of fate, the sense of power and the wonder of the future.

The main part of the book is devoted to chapters on flight, stability and control, rigging, propellers and maintenance. The ideas fancifully discussed in the prologue are here explained at greater length. In the chapter on flight the author describes the factors governing lift-drift ratio, a knowledge of these being an absolute necessity to any one responsible for the rigging of a machine and its maintenance in an efficient condition. The reason for the ever prevailing compromise between climb and velocity is explained by the following summary:

Essentials for maximum climb: low velocity, large surface, large angle relative to propeller thrust, large angle relative to direction of motion, and large camber.

Essentials for maximum velocity: high velocity, small surface, small angle relative to propeller thrust, small angle relative to direction of motion, and small camber.

After considering longitudinal, lateral and directional stability, instability and neutral instability, the author explains the causes of banking, side slipping, spinning, and tells how to make a gliding descent when this is necessitated by a pilot finding himself in a fog, and also the necessary precautions to take in looping the loop and in upside-down flying.

The chapter on rigging sets forth very completely the precautions necessary to take in assembling and adjusting the different parts of the machine so that it is ready

to fly. Directions are given for checking the quality of the wood, the assembly of the different wires, washers, turnbuckles and the adjustment of control cables. The next step is to see that the angle of incidence, the dihedral angle, are correct and that the other parts are truly aligned.

The final chapter on the propeller, or as it is called in England, the "air-screw," is one of the most interesting parts of the book. The theory of the propeller is explained briefly, and conditions to be observed in maintaining its efficiency are outlined. Methods of testing for constant pitch, straightness, length, balance, surface area and camber are explained. The chapter is concluded with a discussion of the relative value of four and two-bladed propellers. The author points out that the four-bladed propeller is only suitable for large pitch, to secure which the propeller must be geared to rotate at a lower speed than would be the case if directly attached to the engine crankshaft.

The final chapter on maintenance takes up the general care of the machine and also goes into the precautions to be taken in the maintenance.

A glossary and a series of plates showing different types of aircraft are given at the end of the book. In the glossary the terms mentioned in the text are defined, the different parts being illustrated.

Throughout the book a number of valuable line drawings are used to illustrate the text. The plates mentioned are of different types of aircraft, showing the development up to the period just before the war. The illustrations represent French and English machines mainly, with one or two American types.

AIRCRAFT IN WAR AND COMMERCE. By W. H. Berry. (With introduction by Col. Lord Montagu of Beaulieu, C. S. I.) Published by Geo. H. Doran Co. Cloth, 5½ by 8¼ in., 123 pp., illustrated. Price \$1.50.

Most of the aircraft books nowadays are one of two types, dealing either with the personal experience of some flyer or with the elements of operating principles, written for flying students. A few engineering books have been published, and their number is increasing, although slowly. Also, a few books on the utilitarian aspects of aircraft have appeared. Mr. Berry's book is a serious discussion of the use of the airplane as a weapon in war, and its possibility as a commercial tool in the future. It gives a wonderfully illuminating view of aeronautic conditions in England.

At the very beginning of the book, the author credits this country with the invention of the airplane, stating that the machine is essentially and wholly American.

The first chapter is an interesting account of the early history of aircraft, tracing its progress through the first flight of the Wright Brothers in this country to the passing of aeronautic supremacy to France, with the work of Farman, Santos-Dumont and others, and concluding with the description of the first flight of Blériot across the English Channel on July 25, 1909, in his monoplane fitted with a three-cylinder, 22-28-hp., air-cooled, Anzani engine.

In the early days there seemed to be little interest taken in the development of aircraft in Great Britain, while at that time France was doing wonderful work. In fact, during the first month of 1914, practically all the flight records in speed, duration, passenger carrying, height and distance were taken by German pilots. At the beginning of the war, the Royal Flying Corps had only about 80 machines that would take the air, not one of which was fitted with a British built engine.

(Continued on page 52, Advertising Section.)